LAWRENCE J. LUKENS

Air-Brake Troubles Brake Rigging

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AIR-BRAKE TROUBLES

FOUNDATION BRAKE RIGGING
Parts 1–2

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FOUNDATION BRAKE RIGGING, PART 1

FOUNDATION BRAKE RIGGING, PART 2

KEY TO COLOR PLATES

Red	*Main-Reservoir Pressure*
Pink	*Brake-Cylinder Pressure*
Dark Green	*Pressure-Chamber Pressure*
Blue Green	*Feed-Valve Pressure*
Light Green	*Equalizing-Reservoir Pressure*
Orange	*Atmospheric Pressure*
Yellow	*Brake-Pipe Pressure*
Blue	*Live Steam*
Purple	*Application-Chamber Pressure*
Gray	*Reducing-Valve Pressure*

AIR-BRAKE TROUBLES

Serial 2513 Edition 1

DISORDERS AND TESTS OF AIR-BRAKE APPARATUS

9½-INCH AND 11-INCH AIR COMPRESSOR

DISORDERS

1. Purpose of Air Compressor.—The purpose of the air compressor is to supply compressed air, without which the brakes cannot be operated; therefore, any disorder that causes the compressor to be inoperative renders the air-brake system useless.

The compressor works under more severe conditions than any of the other operating parts of the automatic brake equipment. Unlike these parts, the compressor is practically in continuous operation, as it must compress the air for charging and recharging the brake system and also keep the brake system charged against leakage. In addition, it must furnish the compressed air for the operation of the auxiliary air-operated devices such as reverse gears, fire-doors, sanders, etc., with which modern locomotives are equipped.

2. Causes of Disorders.—The length of time a compressor will continue to operate satisfactorily depends on the care it receives and the severity of the conditions under which it works. In time, the compressor will develop disorders due to wear and other causes that will reduce its efficiency.

Dirt and improper lubrication are two factors that contribute more than anything else to impair the efficiency of an air com-

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pressor. Dirt if permitted to enter the compressor through the air inlets mixes with the oil to form an abrasive grit that causes excessive wear on the cylinders, pistons, packing rings, and valves. Oil if used too liberally carbonizes under the heat of compression and restricts the passages and this results in an overworked and a quickly worn out compressor. Clean air, which implies a suction strainer in good condition, and oil of the proper quality applied to the air cylinders by a lubricator that feeds the oil in small particles and in an atomized form will insure efficient compressor operation over extended periods.

Compressor disorders are indicated by the stopping of the compressor, irregular strokes, pounding, overheating, blowing, or failure to compress the required amount of air.

When considering the compressor disorders, reference will be made to the diagrammatic views of the 9½-inch compressor shown in Figs. 1 and 2.

3. **Compressor Lubrication.**—The proper lubrication of the air cylinders of the air compressor is a rather difficult matter, more especially with the 8½-inch type. The oil must not be too heavy, or, as it is termed, of a too high viscosity, or it will not atomize properly; also, such an oil will tend to collect any dirt or cinders that may be in the air, and the mixture then bakes on the hot surfaces. In addition, an unsuitable or an improperly refined oil will decompose under the heat of compression and form hard flinty carbon deposits in the cylinders and passages, although an analysis of numerous so-called deposits has shown them to consist more of dirt than of carbon, the whole being held together by the gummy matter from the decomposed oil. The carbon deposits from a properly refined oil will be of a light, fluffy nature.

The best results will be obtained from an oil of a moderately high vaporizing point, a suitable viscosity, and a low carbon content. The oil used in the steam cylinders of a locomotive does not fully meet these conditions and hence is not entirely suitable for the air cylinders of compressors.

4. **Failure of Compressor to Start.**—If the compressor fails to start when the throttle is opened, an investigation should

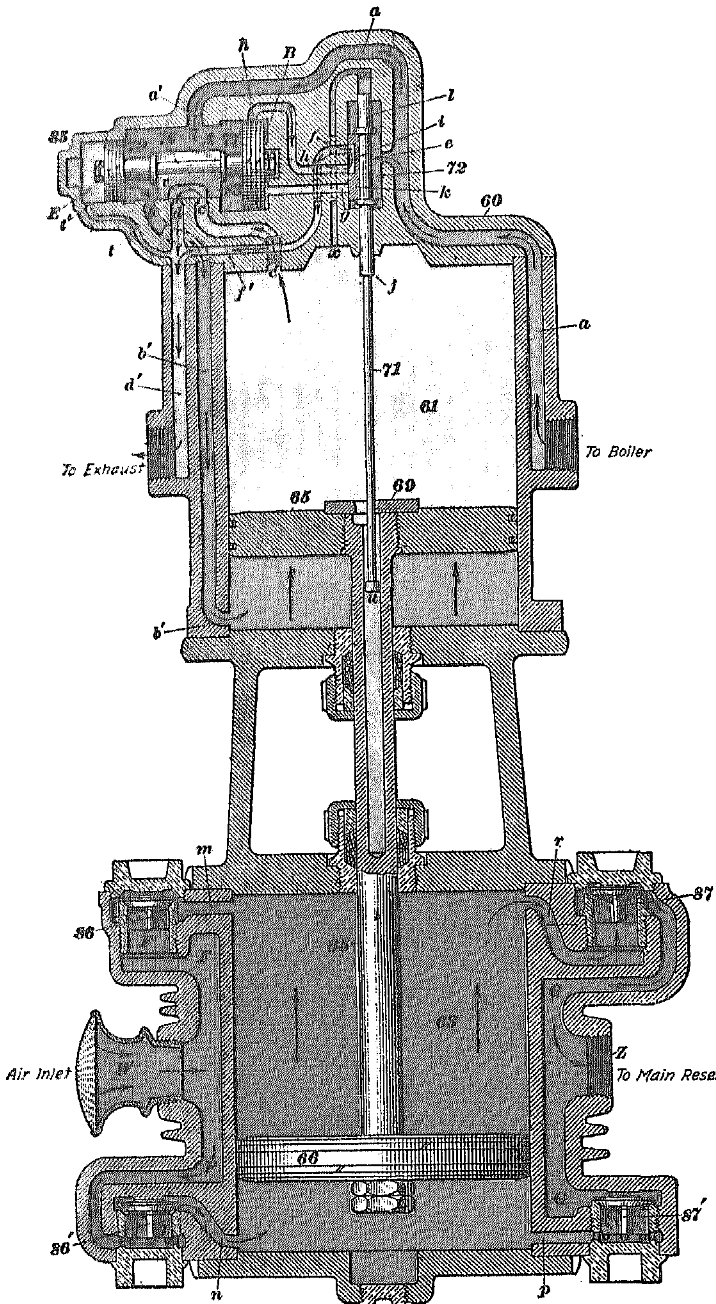

Fig. 1

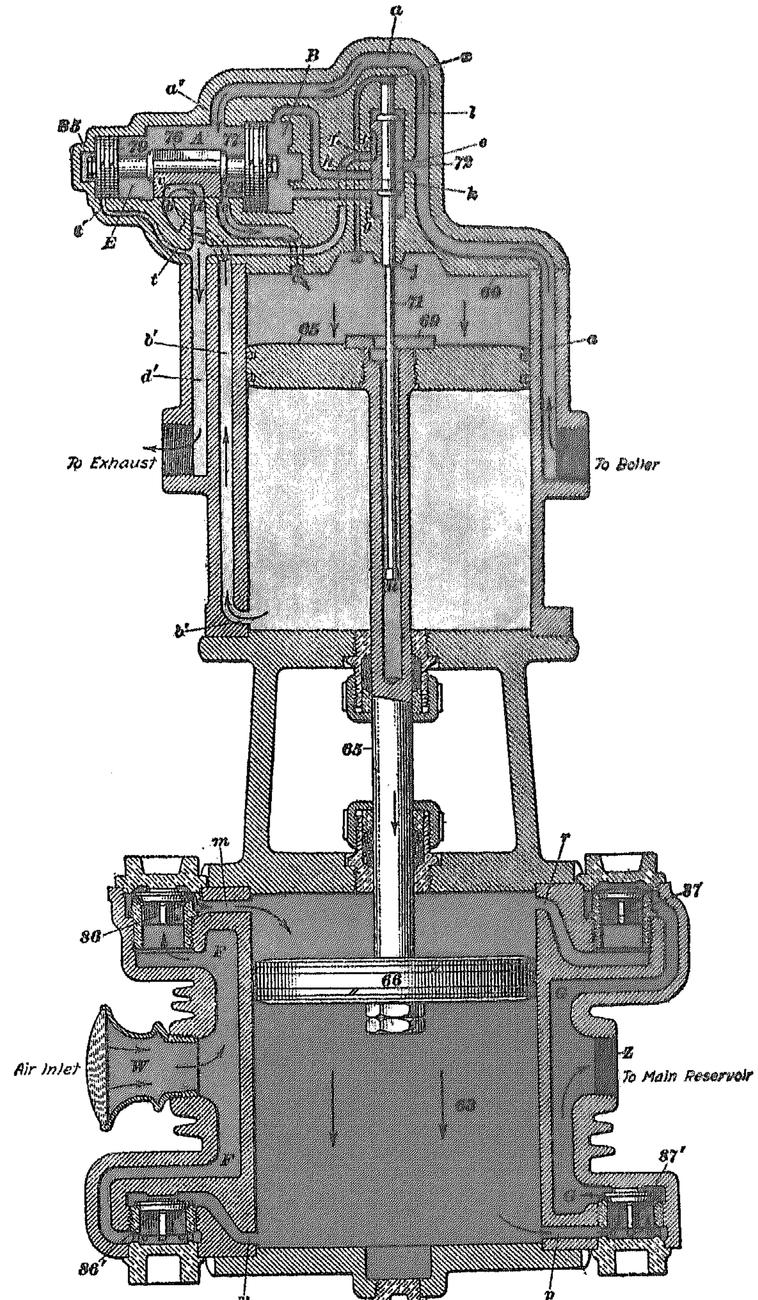

Fig. 2

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be made to see, first, by opening drain cock *106,* not shown, which is placed in the steam passage *a,* whether steam is passing through the governor to the compressor. If a strong blow of steam results, it may be assumed that the trouble is in the compressor; but if not, the steam pipe near the governor should be struck lightly with a hammer in order to jar the governor steam valve open.

With steam in the compressor, its failure to start may be due to the fact that the steam end is not receiving sufficient oil. The packing rings on the differential pistons *77* and *79* may be leaking badly or be stuck in their grooves or an accumulation of rust may be responsible if the compressor has been idle for some time. If the failure to start is due to insufficient lubrication, the compressor can generally be started by closing the steam valve for a few minutes and feeding about twenty drops of oil to the steam end and then opening the valve quickly. This operation will carry the oil to the compressor in the event of its being held up in the piping. If this does not start the compressor, the portion of the steam head containing the main-valve piston should be tapped lightly, thus jarring the piston loose, but the reversing-valve cap should not be struck. If the cap is struck it may be bent to one side and the part of the rod within it cannot move vertically and the rod will probably be bent when the compressor goes to work. Removing the cap nut and pouring in a small quantity of valve oil also assists in starting the compressor. If the compressor still fails to work the trouble is probably due to disorders other than those mentioned.

5. Compressor Stops in Service.—If the compressor stops in service, and will not start after following the procedure outlined in the preceding article, it will generally be found that the cause is due to a failure of some of the parts that reverse the compressor. The nuts may be working loose on the piston rod in the air cylinder, or the packing rings on pistons *77* or *79* may be leaking badly. The failure of the reversing parts to work may be due to a broken reversing-valve rod, or the rod may be disengaged from the reversing plate; the shoulder *j* or

the button *u* on the reversing rod may be badly worn; a reversing plate may be worn; or a plate may be loose owing to the working upwards of the studs that secure it to the piston. All of these disorders prevent any movement from being imparted to the reversing valve as the steam piston completes either stroke. If the compressor cannot be started, the engineer should be guided by the rules of his road governing these conditions. Generally speaking, it is not considered practicable to attempt to make repairs while on a trip.

6. **Leaky Packing Rings on Pistons 77 and 79.**—Leakage by the packing rings on pistons *77* and *79*, if not too bad, is usually indicated by the failure of the compressor to start when steam is turned on first, until the steam head is tapped, provided the compressor is not dry. If the rings do not leak enough to stop the compressor, the compressor will be slow to reverse, as the leakage delays the unbalancing effect on the pistons necessary to move the main valve *76*, and causes it to move slowly toward the leak. If the rings on piston *77* leak, the compressor will be slow in beginning an upward stroke, as the movement to the right of the main-valve piston will be delayed. If the rings leak on piston *79*, the compressor will be slow in reversing after completing its upward stroke, as the leak prevents the main-valve piston from being moved quickly to the left.

If the leak by the rings on piston *79* exceeds the capacity of port *t*, the compressor will stop at the end of its upward stroke, as the pressure on each side of the small piston will be balanced instead of being unbalanced and the main valve *76* will not move the main slide valve to the left.

7. **Compressor Pounds.**—Pounding in the air cylinder, due to the piston striking the head, may be caused by the piston being loose on the rod owing to the omission of the cotter, which causes the nuts to come loose. The clearance in an air compressor is intentionally made as small as possible, the air as it is compressed in the air cylinder being relied on to act as a cushion for the piston and prevent it from striking the cylinder head. The cushioning effect may be destroyed and pounding

caused by leakage past the inlet valves, piston packing rings, or stuffingbox. The compressor will also pound if it is loose on its brackets or if the brackets are loose on the boiler.

In the steam end, a loose reversing plate will make a clicking noise by lifting off the piston and coming back on it again as the compressor reverses to make an up stroke. However, a loose plate is rare because its studs extend through the piston and their ends are riveted over.

Any unusual click or pound in the compressor should be reported promptly, as it will sooner or later lead to a compressor failure.

8. **Compressor Short-Strokes Rapidly.**—The compressor is said to short-stroke when the pistons have an otherwise normal movement but reverse before they make a complete stroke. Short-stroking is caused by the reversing valve moving before the steam piston makes a complete stroke. The compressor when the throttle is first cracked sometimes short-strokes very rapidly or *dances* at the upper end of the cylinder, and as there is no air pressure to cushion the pistons, they strike the cylinder heads. The pounding is sometimes so severe as to jar the compressor on its brackets. The throttle cannot be opened fully on account of possible damage to the compressor, since the pounding increases as the steam supply is increased.

The dancing action is caused by the reversing valve or its seat being somewhat worn. This allows steam to get between the valve and its seat, and decreases the effectiveness of the steam pressure on the back of the valve in holding the valve and the reversing rod in their upward positions, especially if the compressor has been receiving plenty of oil. The action of the steam in holding the reversing valve on its seat is further reduced by the quick downward movement of the pistons on account of their weight, provided there is not too much friction in the stuffingboxes. The steam then expands into the large volume of the cylinder and reduces in pressure. The reversing valve and rod then drop down as soon as the reversing plate moves away from the shoulder of the reversing rod, and the compressor reverses.

The compressor should be allowed to work if it shows any tendency to compress the air, until sufficient pressure has accumulated in the main reservoir to cushion the air piston. If the compressor throttle is now opened fully, the short-stroking will usually stop because the higher steam pressure in the reversing-valve chamber will prevent the reversing valve from dropping down.

The temporary remedy for short-stroking when it occurs on the road is to feed less oil to the steam end of the compressor.

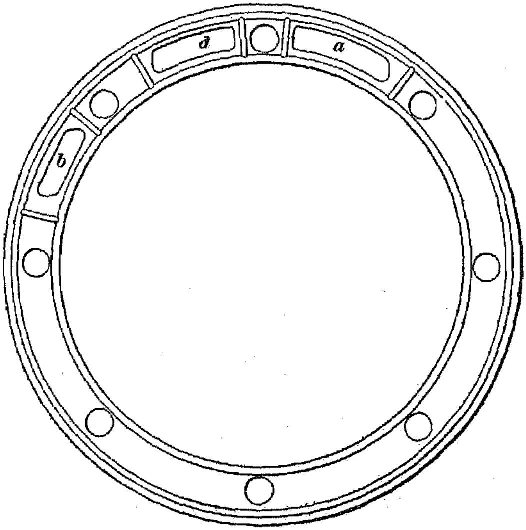


FIG. 3

9. Compressor Groans.—The compressor may groan if the steam or air cylinder is insufficiently lubricated, or because the latter cylinder is dry on account of overheating. Another reason is that the packing in the stuffingboxes may become so hard as to bind on the piston rod. The packing should be renewed and the swab on the rod kept well oiled with valve oil. The steam and air cylinders should receive additional lubrication.

When hand oilers are used on the air cylinder, it is sometimes impossible to introduce oil into the cylinder on account of the

fact that the oil passage becomes filled with hard gum. The oil passage leads from the oil cup diagonally through the center piece toward the center of the air cylinder and is therefore difficult to clean out when it becomes stopped up.

10. Steam Blows.—The principal blows that occur in the steam end of a compressor may be due to: (1) leaks in the copper gasket between the top head and the cylinder; (2) leaky packing rings on the steam piston *65*; (3) worn main slide valve *83* or worn seat.

1. Leaks in the Copper Gasket.—Reference will be made to the top view of the copper gasket shown in Fig. 3, and also to Figs. 1 and 2, when considering the effects of leaks in the gasket. The gasket, Fig. 3, usually breaks into the upper end of the cylinder either from the main steam passage at *a* or the exhaust passage at *d*, or from the steam passage at *b*. As the gasket rarely breaks between the ports, the effect of leakage around the gasket from one port to the other will not be considered. A break in the gasket at *a* that permits steam from the main steam passage to enter the upper end of the cylinder will cause a blow when the piston is making an upward stroke (Fig. 1), as the main slide valve now has the the upper end of the steam cylinder connected to the exhaust. A bad leak at this point in the gasket will cause the compressor to make a slow upward stroke or to stop, depending on the pressure against which the air piston is working. A leak through the gasket at *d* between the upper end of the cylinder and the exhaust passage causes a blow on the downward stroke, as the upper end of the cylinder then contains live steam, and this stroke will be slower than the upward stroke. A leak through the gasket at *b* from the upper end of the cylinder into the steam passage to the lower end of the cylinder causes a constant blow, as on the downward stroke, Fig. 2, live steam leaks into passage *b* and escapes at the exhaust, and on the upward stroke, Fig. 1, part of the steam that is passing to the lower end of the cylinder escapes through into the upper end of the cylinder and thence to the exhaust. The effect of these leaks will be to stop the compressor or reduce its speed.

2. *Leaky Packing Rings on Steam Piston.*—Leakage past the packing rings on the steam piston will cause a constant blow, and will also cause the compressor to run slowly when working against any considerable pressure.

3. *Leaky Main Slide Valve.*—A leak past the main slide valve *83* will cause a blow on either or both strokes, depending on the part of the valve or valve seat that is worn.

TEMPERATURE OF AIR COMPRESSOR

OVERHEATING

11. Conditions Affecting Temperature.—The temperature of the air cylinder depends upon: (1) the speed of the compressor; (2) the pressure to which the air is compressed; (3) the temperature of the air before compression; (4) the condition of the air compressor. The heating of air during compression cannot be prevented and is due to the friction encountered by particles of the air as they are forced over each other when crowded together and compressed in the air cylinder. This action may be compared to the heat produced by pressing two smooth blocks of wood firmly together and then moving one over the other, the degree of heat evolved depending on the pressure and the speed of the moving block.

12. A distinction should be made between the normal heating of air that always accompanies compression, and the abnormal temperature resulting from causes that interfere with the compression of air. The following shows the normal heat which results when the air compressor is in good condition. With the temperature of the incoming air at 70° F., and the compressor working at a speed of 140 exhausts or strokes per minute against a main-reservoir pressure of 130 pounds, the final temperature of the air was found to be about 550° F. If the speed of the compressor and the main-reservoir pressure had been less, the final temperature would have been lower.

Overheating then will result from the normal heat of compression alone when the compressor is in good condition and run at about maximum speed for any considerable length of time against a high main-reservoir pressure. The compressor,

therefore, must not be run at a continuous high speed, and if it is necessary to do so the indications are that there is excessive leakage somewhere in the brake system, or that the supply of compressed air required is beyond the capacity of the compressor.

13. **Overheating.**—Overheating decreases the efficiency of the compressor, as the disorders that produce overheating generally result in less air being drawn in to be compressed. Overheating also burns out fibrous packing or melts and blows out metallic packing, and destroys the lubrication, thereby increasing the wear on the cylinder. The compressor then should be maintained and operated so as to prevent the heating of the air to a greater degree than that due to the normal amount of compression.

14. **Causes of Overheating.**—Overheating is generally due to disorders in the compressor that cause the air to be compressed over and over again and thus heated to a much higher temperature. If the air cylinder is properly lubricated and the compressor is not run continuously at high speed against a high pressure, the temperature of the air cylinder in excess of the normal heat attending compression depends upon the condition of the air valves and the packing rings on the air piston. The operation of the air valves is affected by dirt that enters with the air and combines with the oil and worn-off metal to form a gum which hardens with the heat. The gum adheres to the air valves and causes them to leak, or stick, or it restricts their lift. The principal causes for the overheating of the compressor, the most of which also cause the compressor to make irregular strokes, are as follows: (1) Leaky air-piston packing rings or a badly worn air cylinder; (2) discharge valves that leak or have improper lift, or restricted discharge passages; (3) inlet valves that leak or have insufficient lift, or obstructed air inlet passages; (4) leakage at the stuffingbox.

1. Leaky Air-Piston Packing Rings, or Worn Air Cylinder. Probably the most common cause of excessive heating is leaky air-piston packing rings or a worn air cylinder. With these

defects the air can leak by the piston in either direction. Consequently, less air is taken into the air cylinder and less air is forced into the main reservoir. This causes the compressor to make a greater number of strokes in compressing a given amount of air. The air that leaks past the piston has had its temperature raised by compression, and therefore the temperature of the incoming air will be raised, resulting in a still higher temperature.

2. *Leaky Discharge Valves, Valves With Insufficient Lift, or Restricted Air Passages.*—Leaky discharge valves and packing rings cause excessive heating for the same reason, as the leaky valves permit the hot air in the discharge pipe to return to the air cylinder where it is again compressed and further heated. The leakage of air also prevents the air from being drawn into the air cylinder, thereby causing the compressor to work more rapidly, and prevent the cooling of the cylinder. Discharge valves with insufficient lift or restricted air passages hold back the discharge of air to the main reservoir and cause overheating. This is due to the much higher pressure developed in the air cylinder before the air is discharged to the main reservoir, and to the greater friction encountered by the air in passing through the restricted passages. The leakage by the air-piston packing rings is also increased and the normal cooling of the air cylinder is prevented, because less air is drawn in.

Leaky discharge valves are often caused by an accumulation of carbon that follows the use of too much oil, some of which carbonizes under the heat of compression. The carbon is found mostly at the bottom discharge valve.

3. *Leaky Inlet Valves, or Inlet Valves With Insufficient Lift.*—Inlet valves that leak, or that have insufficient lift, or air passages that are obstructed cause overheating on account of the additional speed of the compressor necessary to compress a given amount of air.

4. *Leakage at the Stuffingbox.*—Stuffingbox leakage causes overheating in two ways: (*a*) By the additional speed necessary to compress the required amount of air; and (*b*) by the friction and consequent heating of the air caused by the leak. This leak also impairs the efficiency of one end of the cylinder.

15. **Cooling an Overheated Air Cylinder.**—About all that can be done when the air cylinder becomes overheated is to increase the lubrication. The trouble should be reported at the end of the trip.

TESTING THE AIR CYLINDER

16. **Order in Which the Test Should Be Made.**—The following order should be observed when testing the air cylinder: First test the discharge valves; then the packing rings on the air piston; and last the inlet valves. When the test is made in this order a disorder in any one of the above parts will not interfere with the testing of the others.

17. **Testing the Discharge Valves.**—To test the discharge valves, the main-reservoir pressure should be increased to maximum, and the compressor stopped. Then open the oil cup, and remove the plug in the lower cylinder head. A leak at the upper discharge valve will cause a blow at the oil cup, and if the lower discharge valve leaks, a blow will occur at the plug opening. Leakage too slight to cause an audible blow, can be found by a light, or by holding the finger or hand tightly over the opening and then removing it. A leaky discharge valve is generally indicated by the pale brick-red color of the boss or projection in which the valve is located, this discoloration being due to the heat which is produced by the defective valve.

18. **Testing Air-Piston Packing Rings.**—To test the packing rings on the air-piston, replace the plug, and run the compressor at about thirty exhausts per minute against a pressure somewhat less than that at which the governor operates, and note whether any air escapes at the oil cup on the downward stroke. If the rings are in good condition, the piston should complete the stroke without any evidence of a blow. If it is desired to test the rings on the other stroke, the oil cup may be closed and the plug in the lower head removed, and the indications of leakage observed on the upward stroke. If the rings are defective the extent of the leakage past them may be judged by noting at what point of the stroke the blow begins.

When swab holders are applied, it is difficult to tell one stroke from the other. However, the downward stroke may be

detected, as it is always slightly faster, because the action of the steam is assisted by the weight of the piston and the piston rod.

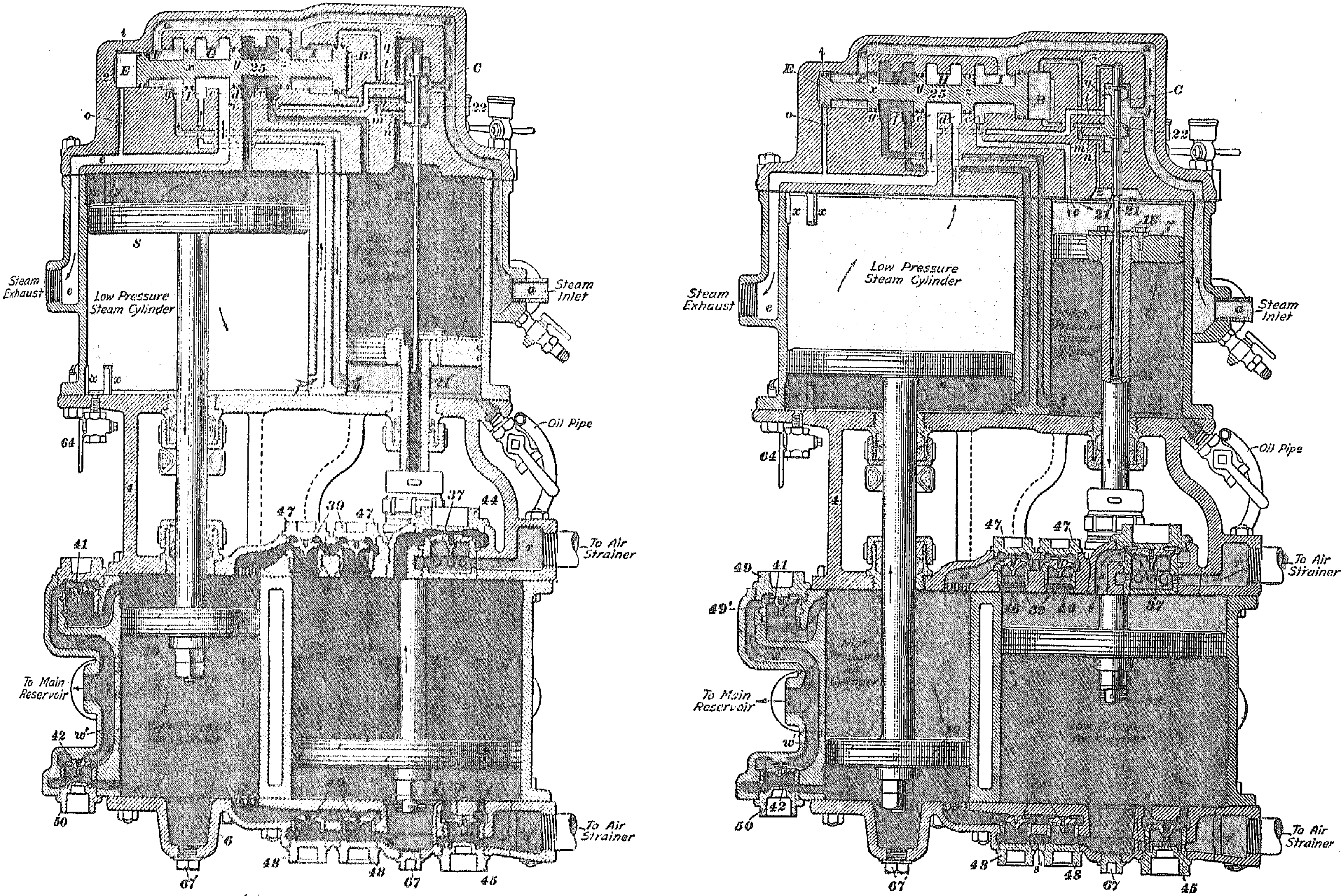
Leaky packing rings are always indicated by the diminished supply of air taken in at the strainer; there will be practically no air taken in on the last half of the stroke. An indication of leaky rings is a dull thud as the compressor reverses. The thud will turn into a pound as ring wear increases.

19. **Testing the Inlet Valves.**—A leaky inlet valve permits the air that is escaping to pass to the other end of the cylinder, instead of blowing back through the strainer, thereby causing less air to be drawn in. The best test for a leaky inlet valve is to listen at the strainer for the sound made by the air when it escapes past the leaky valve.

REMOVING THE REVERSING ROD

20. To remove the reversing rod, take out the plug in the lower cylinder head, and with a bolt, push the air piston up to about the middle of its stroke, first opening the oil cup to relieve the air pressure above the piston. Remove the reversing-valve-chamber cap and pull the reversing rod upwards until the button on the rod strikes the reversing plate. The larger part of the reversing rod is now completely out of its bushing, and as the lower part of the rod is smaller than the bushing through which the larger upper part of the rod operates, the rod can now be worked sideways until the button on the rod is pulled through the offset hole in the reversing plate. To prevent bending the reversing rod, it must always be taken out before the steam cylinder head is removed. The steam throttle must be closed tightly and all drain cocks on the steam cylinder opened before attempting to remove the reversing rod.

Another method when soft packing is used and the compressor has steam, is to run it very slowly, and then tighten the packing nut when the piston is about halfway in the cylinder.



(a) (b)

FIG. 4

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8½-INCH CROSS-COMPOUND COMPRESSOR

DISORDERS

21. When considering the disorders of the 8½-inch compressor, reference will be made to the diagrammatic views of the compressor shown in Fig. 4, (*a*) and (*b*). The disorders that develop in the steam end of the 9½-inch or 11-inch air compressors and cause them to stop or operate improperly will affect the 8½-inch cross-compound compressor in the same manner, as the reversing mechanism of all three types of compressors is similar. It is therefore unnecessary to consider the disorders which affect the reversing mechanism of the compressor.

The fact that the steam is used twice, or compounded, thereby requiring a low-pressure steam cylinder, does not render the steam portion liable to additional failures, as the operation of the low-pressure steam piston is accomplished without the addition of any extra parts. However, as the 8½-inch compressor compresses the air twice and therefore requires another air cylinder, piston, and additional air valves, disorders in these valves or in the high-pressure air piston packing rings affects the operation of this type of compressor differently than single-stage compressors in which the air undergoes only one stage of compression. It is, therefore, possible to confine this Section largely to disorders which involve the air cylinders alone.

22. Compressor Pounds.—The principal causes of pounds in the compressor are leaky packing rings on the high-pressure piston and leaky intermediate air valves. Both of these disorders destroy the cushioning effect of the air on the high-pressure piston and cause it to strike the head of the cylinder. Owing to the weight of the pistons the pound will be more pronounced on the down stroke.

The pounding may not be especially noticeable when the compressor is running at speed but, when running slow, as when the governor is about shutting off, the piston when near the end of its stroke seems to slip and strike a severe hammer blow on the cylinder head.

Usually but little trouble is experienced with pounding in the low-pressure air cylinder. The packing rings wear much less rapidly on the low-pressure air piston; also, the air is compressed only to about 40 pounds, and the effect of a lessened air cushion will not be so apparent as in the other cylinder.

23. **Compressor Makes Irregular Strokes.**—The compressor makes irregular strokes because of leaky or stuck air valves, or restricted air passages between the air cylinders. Irregular strokes are always accompanied by overheating.

24. **Compressor Overheating.**—The principal causes of overheating as well as a failure to compress the required amount of air are leaky packing rings on the air pistons, leaky air valves, air valves stuck open or with improper lift, obstructed air passages or oil cups stopped up.

The heating generally occurs in the high-pressure air cylinder owing to the high-pressure air piston operating against air of a high temperature; the low-pressure air cylinder receives air from the atmosphere. The high-pressure cylinder is also more difficult to lubricate not only because of the higher temperature but also because of the short time during which lubrication can be applied. There is only a short interval on the down stroke of the high-pressure air piston when oil can feed by gravity to the cylinder before the compression of the air following the upward stroke of the low-pressure piston stops the flow of oil.

25. **Compressor Runs Slow.**—The disorders that cause the compressor to run slow and fail to compress enough air are leaky packing rings on the steam pistons, leaky air-piston packing rings, particularly on the high-pressure air piston, leakage across the upper or lower steam cylinder gasket; low steam pressure or wet steam caused by a disconnected or leaky compressor dry pipe, as when the compressor is not operated by superheated steam, and leaky intermediate or discharge air valves or valves with restricted lift. Carbon that develops owing to some of the oil carbonizing under the heat of compression is generally the cause for leaky valves or for valves with impaired lift. The bottom discharge valve is the one most generally affected.

Leaky packing rings on the steam pistons, or worn pistons will be noticed by the slapping of the pistons against the cylinder walls as well as by a chugging or muffled sound while the compressor is working. Leaky rings on the low-pressure steam piston will cause a blow at the stack; leaky rings on the high-pressure piston will slow up the movement. The rings wear much more rapidly on the high-pressure than on the low-pressure steam piston.

The compressor will have to be removed from the engine and dismantled to examine the steam-cylinder gaskets. They will be found to leak straight across between the cylinders and not between the ports. This leak will cause the compressor to run slow when the air pressure reaches 75 or 80 pounds.

26. Compressor Stops.—The disorders which will develop in the steam end of the $8\frac{1}{2}$-inch compressor and which will cause it to stop are the same as already mentioned in connection with the $9\frac{1}{2}$-inch and 11-inch compressors. Another disorder that will stop the compressor is found in the air cylinder, and is due to a discharge valve that is stuck open or broken, or to a loose valve seat. This allows main-reservoir air to enter the high-pressure air cylinder, and will stop the compressor when a main-reservoir pressure of about 40 pounds has been reached.

27. Reason Why the Compressor Stops at 40 Pounds. The compressor stops with a main-reservoir pressure of 40 pounds when a discharge valve sticks open or leaks, because the load on the low-pressure air piston is now equal to the force which is exerted by the steam on the high-pressure steam piston. The following calculation will show that the steam and air pressures on these pistons become balanced with an air pressure of 40 pounds:

A boiler pressure of 200 pounds gives a working pressure of about 190 pounds to the square inch in the high-pressure steam cylinder, but as the steam in the low-pressure steam cylinder exerts a back pressure of about 70 pounds to the square inch on the high-pressure steam piston, an effective pressure of 120 pounds to the square inch is left to operate the high-pressure steam piston. The diameter of the high-pressure

steam piston being 8½ inches, its area is 56.7 square inches, and when this area is multiplied by 120 pounds, the result arrived at is about 6,800 pounds, the force exerted by the high-pressure steam piston upon the low-pressure air piston.

The diameter of the low-pressure air piston is 14½ inches, and its area is then 165 square inches. Dividing the force exerted by the high-pressure steam piston, or 6,800 pounds, by 165 gives the maximum air pressure in the low-pressure cylinder against which the high-pressure air piston will work, or about 40 pounds.

When a discharge valve sticks open or breaks,, the intermediate valves in the same end of the cylinder will be held closed by a pressure of 40 pounds to the square inch or more, depending on whether the compressor has just been started or has been operating. The compressor will, therefore, stop when the air in the low-pressure air cylinder is compressed to about 40 pounds.

TESTING THE STEAM ENDS OF COMPRESSORS

28. Method of Test.—The method used in testing the steam ends of the various compressors is to observe the speed they develop with a certain boiler pressure in maintaining the main-reservoir pressure at a certain fixed pressure.

The chart shown in Fig. 5 has speed curves of the 9½-inch, 11-inch, and 8½-inch compressors plotted on it, *ef* being the speed curve of the 9½-inch compressor, *cd* being the speed curve of the 11-inch compressor, and *ab* being the speed curve of the 8½-inch compressor. This chart and the test specifications given later on were prepared by the Westinghouse Air Brake Company.

29. Explanation of Chart.—Reading from the left to the right, the chart, Fig. 5, is divided by heavy vertical lines. Each one shows an increase in steam pressure of 10 pounds. The light vertical lines divide the distance between two of the heavy lines into ten equal spaces, each space, therefore, indicating an increase in pressure of one pound.

Reading from the bottom to the top, the chart is divided into equal spaces by heavy horizontal lines. Each one, with the

exception of the second, is marked as showing an increase of 20 strokes per minute. The light horizontal lines, with the exception of the first ten, divide the distance between two of the heavy lines into ten equal spaces, each space representing an increase in speed of two strokes per minute.

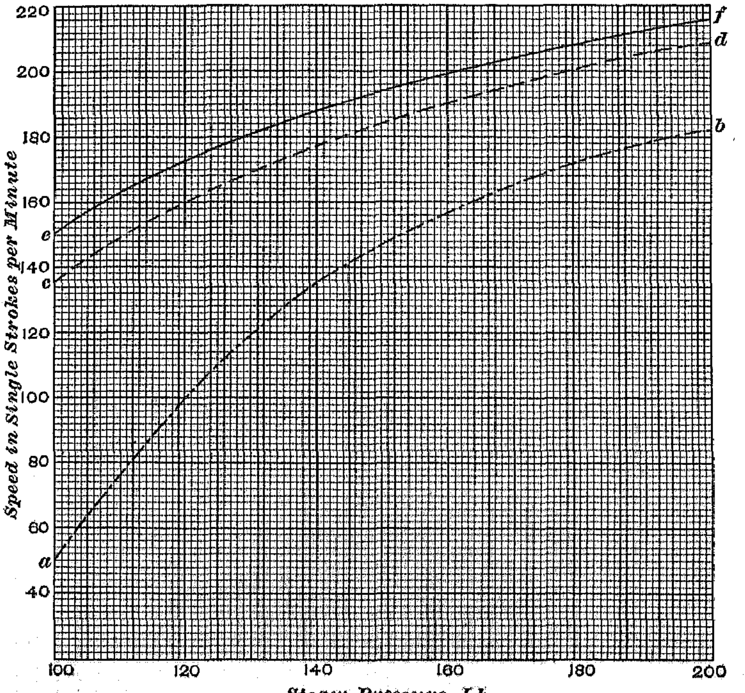

FIG. 5

30. **Use of Chart.**—To use the chart, find the vertical line corresponding to the boiler pressure carried. Follow this line upwards until it intersects with the speed curve of the compressor being tested. Then follow the horizontal line found at the point of intersection of the vertical line with the speed curve, to the left, and observe the number of strokes per minute it represents. This latter figure should be the speed of the compressor under the conditions given later on in this lesson paper.

31. **Making the Test.**—It will be assumed that the steam end of the 8½-inch cross-compound compressor is to be tested. To make the test, open the compressor throttle wide, and regulate the main-reservoir pressure by making a leak so that the compressor will maintain a pressure of 53 pounds. If the boiler pressure is 120 pounds, the chart shows that the compressor should make 100 exhausts per minute if in fairly good condition.

Whether the compressor will be overhauled in the event of a lesser speed than this, will depend on the rules established for the test. If the condemning limit has been set at 75 per cent. of the average performance, a speed of 75 strokes per minute or less will require that the compressor be repaired. The condemning limit of the compressor should not be set below 75 per cent. of the specified test, because the compressor would then get into such poor condition as to require heavy repairs.

The 9½-inch and 11-inch compressors are tested in the same manner, with the exception that the main-reservoir pressure is maintained at 59 pounds with the former compressor and 66 pounds with the latter.

TESTING THE AIR CYLINDER

32. **Testing the Discharge Valves.**—To test the discharge valves, the main-reservoir pressure should be increased to the maximum pressure carried, and the compressor stopped. To test the upper discharge valve, open the oil cup, or disconnect the oil pipe at the cylinder, whichever is the more convenient. To test the bottom discharge valve, remove the plug in the bottom of the high-pressure cylinder head. The valves can then be tested in the same manner as the 9½-inch compressor.

33. **Testing Packing Rings on the High-Pressure Air Piston.**—The test for a leak past the air-piston packing rings, as made with 9½-inch compressor, will not apply when testing the packing rings on the high-pressure air piston, because the air that is being discharged from the low-pressure air cylinder will interfere with the test. This discharge of air could be prevented by removing the inlet valves, but if this were done, the

compressor could not be run fast enough to make the test reliable, because the low-pressure air piston would then have no air cushion, and severe pounding, with possible damage to the compressor, would result.

The best indication of leakage by the packing rings on the high-pressure air piston is overheating as indicated by the metal around the cages and the discharge pipe becoming discolored, and also by pounding.

34. **Testing the Intermediate Valves.**—So little trouble is experienced with the intermediate valves that no test is required. However, they can be tested as follows: To test the upper intermediate valves, the oil pipe should be disconnected from the low-pressure air cylinder, and the plug removed from the bottom of the cylinder. Leakage by the upper intermediate valves sufficient to interfere with compressor operation will be indicated by the escaping of air at the oil hole when the high-pressure air piston is making an upward stroke. Leakage by the lower intermediate valves will be similarly indicated at the plug opening when the high-pressure air piston is making a downward stroke.

35. **Testing Packing Rings on Low-Pressure Air Piston.** The packing rings on the low-pressure air piston can be tested in the same manner as the $9\frac{1}{2}$-inch compressor.

36. **Testing Inlet Valves.**—If the inlet valves have separate air inlets, the valves can be tested by noting if any air passes back through the valves when the low-pressure air piston is moving toward them. When the air strainer *54* is used, it should be removed, and each opening tested.

CARE OF COMPRESSOR

CLEANING THE AIR CYLINDERS

37. **The Compressor Laundry.**—The air cylinders of compressors are cleaned by circulating a solution of hot lye through them. The apparatus for doing this work is called a *compressor laundry*. This apparatus consists of a tank for the lye solution, a steam coil to keep the solution hot, and suitable pipe connec-

tions, so that the tank can be connected to the inlet and discharge openings of the compressor. The solution is made by dissolving concentrated lye in water, the proportion being 1 pound of lye to 1 gallon of water. This work is done when the locomotive goes to the back shop for general repairs.

38. **Using the Laundry.**—Remove the air strainer from the compressor, connect the supply pipe in the tank to the opening, connect the return pipe in the tank to the discharge opening, and connect the steam line to the steam coil. The air cylinder piston-rod packings should, of course, be tight.

Turn the steam into the steam coil, and when the lye solution is at or near the boiling point, start the compressor, and run it slowly for a few hours. The solution will be drawn from the tank into the air cylinders through the supply pipe, and discharged into the tank through the return pipe. After the compressor has been thoroughly cleaned, the pipes should be disconnected and clean water worked through the compressor for several minutes, in order to remove thoroughly the lye solution. The compressor should then be run slowly until every trace of water has been removed, after which the cylinders should be well lubricated.

TYPE S STEAM COMPRESSOR GOVERNOR

DISORDERS

39. Reference will be made to the Type S steam compressor governor shown in Fig. 6, when considering steam compressor governor disorders, as similar disorders will affect all types of governors in the same way.

The steam compressor governor when in good condition should stop the compressor at the predetermined main-reservoir pressure, and should start it again when the pressure has been reduced not more than 2 pounds below the predetermined pressure. At other times, the governor should permit of normal compressor operation. The governor stops the compressor by admitting air at a pressure determined by the tension of the regulating spring *41*, above steam-valve piston *28*, and starts it by exhausting the air from it. Therefore, disorders that result

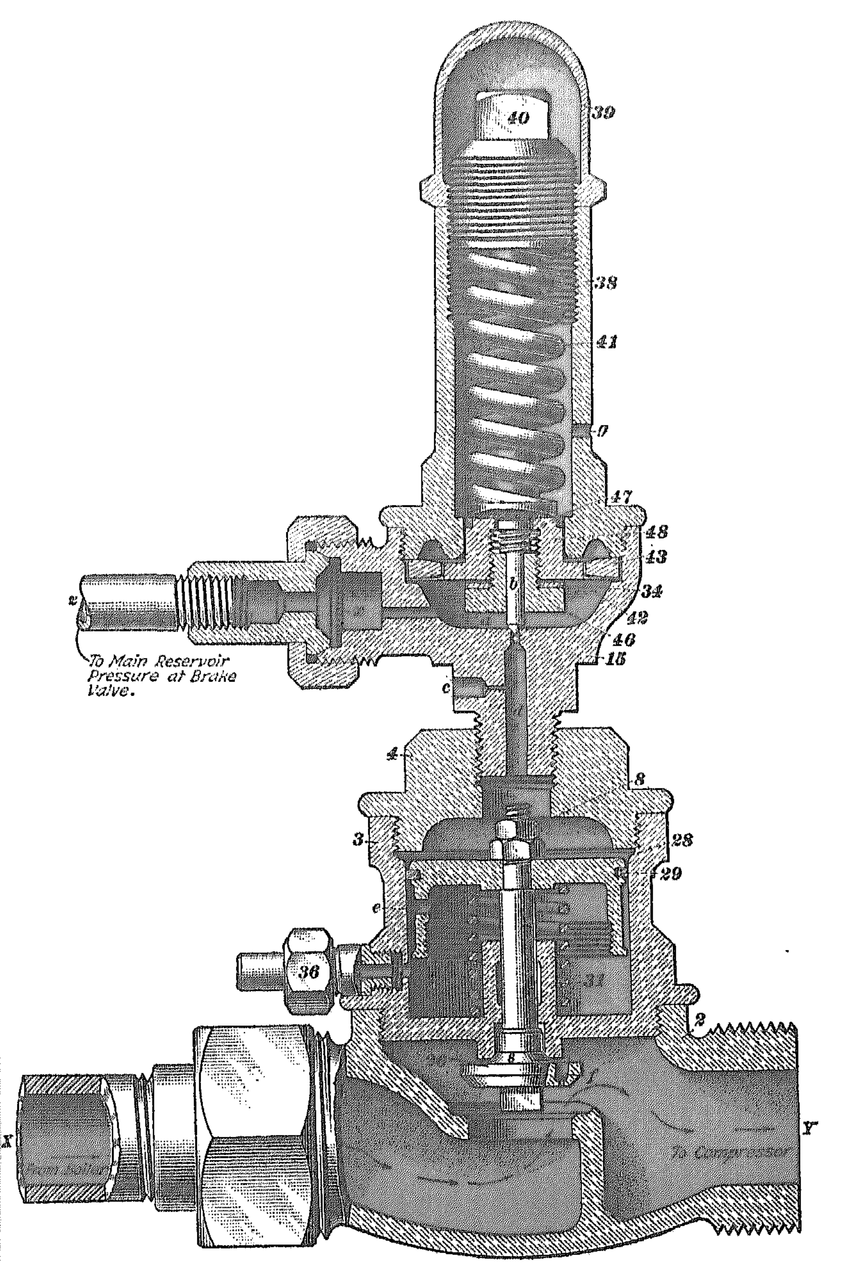

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Fig. 6

in the premature admission of air above piston *28*, or that delay its exhaust therefrom, will cause the governor to interfere with the normal operation of the compressor.

40. Division of Disorders.—The disorders that result in improper operation of the governor may be divided into two classes: (*a*) Disorders that prevent the governor from stopping the compressor promptly when the main-reservoir pressure desired has been obtained; and (*b*) disorders that prevent the governor from starting the compressor promptly after the main-reservoir pressure has fallen 1 or 2 pounds, or that stop or slow it down before the desired pressure has been obtained. It is assumed when governor disorders are considered that the adjustment of the regulating spring is such as to give the desired pressure.

41. Failure to Stop Compressor.—The most common causes for the failure of the governor to stop the compressor at the desired pressure by preventing the steam valve from closing, are as follows: (1) Piston *28* stuck in its cylinder with the steam valve open; (2) packing ring *29* badly worn; (3) port *d* stopped up; (4) a cracked diaphragm *42*; (5) steam valve, seat, or stem badly worn. Less common causes are a stopped-up waste port, or waste pipe when used, the spring *47* may be missing, or the port leading into chamber *a* may be stopped up.

1. *Piston 28 Stuck in Cylinder With Steam Valve Open.* Piston *28* is sometimes prevented from moving freely by the accumulation of a hard deposit of dirt and oil on the wall of the cylinder. Bruising the cylinder by striking it with a hammer may also cause a piston to stick. When the piston sticks, the diaphragm valve will unseat when the predetermined main-reservoir pressure is obtained, but the air admitted above the piston will not be sufficient to force the piston downwards. The main-reservoir pressure obtained before the piston moves and closes the steam valve, will depend on how badly the piston is stuck. This defect is indicated by a blow at the relief port *c* and also by the fact that the compressor is operating when the main-reservoir pressure is above standard.

2. *Badly Worn Packing Ring 29.*—When the packing ring *29* is badly worn, the air which is admitted above the piston when the diaphragm valve first unseats will leak by the packing ring without moving the piston. The compressor, therefore, continues to operate until the supply of air due to the higher pressure is in excess of that passing the packing ring, when the accumulation of pressure will cause the piston to seat the steam valve. A very bad ring, however, will prevent the piston from closing the steam valve.

This disorder is indicated by a blow of air at the relief port *c*, and at the waste port, or pipe, with the compressor operating and the main-reservoir pressure above the predetermined amount.

3. *Stopped-Up Port d.*—Port *d* is sometimes found closed by dirt and oil that is so hard that it has to be drilled out. This defect renders the governor wholly inoperative as it prevents air from entering above piston *28* when the diaphragm valve unseats. The compressor will continue to operate unless stopped by closing the compressor throttle until the pressure in the steam cylinder is unable to move the steam piston against the air pressure in the main reservoir and air cylinder.

4. *Cracked Diaphragm 42.*—A cracked diaphragm will prevent the diaphragm valve from lifting and stopping the compressor until the pressure in the diaphragm chamber becomes greater than the crack can take care of. The diaphragm then raises the pin valve and the compressor stops. The extent that the pressure in the main reservoir exceeds standard will depend upon the leak. This disorder is indicated by a blow at port *g* in the spring box increasing in intensity as the pressure is being raised.

5. *Badly Worn Steam Valve, Seat, or Stem.*—The frequent seating of the steam valve *26* wears the face of the steam valve and its seat until in time the valve is not steam-tight. The compressor then keeps operating at a speed depending on the extent of the steam leak after standard pressure has been obtained. A leak by the stem of the steam valve permits steam to enter beneath piston *28*, but the leak will not interfere with the operation of the piston unless the inflow of steam is greater

than the waste port can take care of. In this event the piston will be prevented from moving fully downwards until the main-reservoir pressure increases sufficiently above standard to overcome the force exerted upwards by the steam. This leak is indicated by a blow of steam at the waste port when the compressor is operating with the main-reservoir pressure above the predetermined amount, as the upper beveled shoulder *s* on the the steam valve is now moved away from its seat and allows steam to pass by the worn stem. The beveled shoulder *s*, when the steam valve is fully open, prevents leakage of steam under the steam-valve piston, and any leakage at the waste port under this condition is due to the fact that the shoulder is not making a steam-tight joint with its seat.

If spring *47* is omitted or if it is too weak to hold the boss on the diaphragm valve up against he diaphragm nut, the diaphragm will raise without lifting the diaphragm valve. If the waste port or waste pipe *36* is frozen or stopped up, any leakage of steam or air under the piston cannot escape. The result is that the pressure on the piston will become balanced and the spring *31* will hold the steam valve open. These disorders, as well as a stopped-up strainer in the air pipe, or the port leading to chamber *a* render the governor inoperative.

42. Failure to Start Compressor Promptly.—The most common disorders that prevent the governor from starting the compressor promptly, or that stop it or slow it down before the desired pressure has been obtained are: (1) Stopped-up vent port *c*; (2) leaky diaphragm valve; (3) piston *28* stuck in its cylinder with steam valve closed.

1. Stopped-Up Vent Port c.—The purpose of the vent port *c* is to allow the air to escape from the chamber above the piston chamber when the diaphragm valve closes, in order that the compressor may start promptly. Port *c* also permits any slight leakage past the diaphragm to escape. If port *c* is stopped up, the air cannot escape when the diaphragm valve closes, and it will have to leak past the packing ring *29* and out of the waste port. The rapidity with which it will do this depends on the packing ring; if it fits snugly, the steam valve *26* may not open

until some time after the diaphragm valve closes. In this case the leaks will reduce the main-reservoir pressure considerably before the compressor starts to work again. The inner part of port *c* is very small, but it can usually be opened by means of a needle or pin.

A stopped-up port *c* is sometimes indicated by the fact that the compressor throws water over the jacket when the engine is standing. The reason is that considerable condensation will take place in a compressor when it is prevented from starting by an obstruction in port *c*, especially if the port *f* in the steam valve is also closed.

2. Leaky Diaphragm Valve.—The diaphragm valve rarely leaks owing to the heavy pressure exerted on it by the diaphragm spring. Such a leak will be indicated by a blow of air at the vent port when the compressor is working and will cause the governor to be somewhat less prompt in starting the compressor.

3. Piston 28 Stuck With Steam Valve Closed.—If piston *28* sticks in its cylinder with the steam valve closed, thereby stopping the compressor, it can generally be jarred open by striking the steam pipe near the governor with a hammer.

NECESSITY OF CORRECT LENGTH OF DIAPHRAGM VALVE

43. The sensitiveness of the governor in starting the compressor is largely dependent on the diaphragm valve *b* being maintained at the correct length. The valve may become too short either by wear or grinding. In this event, the governor will stop the compressor at the predetermined pressure, but the governor will be slow in starting the compressor because the regulating spring *41* has to overcome additional friction by crimping the diaphragms when seating the shorter valve. A very slight shortening of the valve will cause a fall in pressure of several pounds before the compressor will start.

DISTORTED DIAPHRAGM

44. A distorted diaphragm *42* will cause erratic operation of the governor, as at times it will stop the compressor above the predetermined pressure, and at other times below it.

NUTS OFF STEAM-VALVE STEM

45. The nuts *8* sometimes work completely off the stem of the steam valve. The operation of the governor is not materially affected thereby and the absence of the nuts will be indicated by a slight blow of steam at the vent port *c* when the governor opens, because of leakage by the steam-valve stem before the steam valve seats upward.

CUTTING OUT A GOVERNOR

46. The governor can be cut out by placing a blind gasket in the air-pipe connection to it. The compressor must then be controlled by the compressor throttle until repairs are made.

TYPE AD STEAM COMPRESSOR GOVERNOR

DISORDERS

47. The type AD steam compressor governor, Fig. 7, is, in general, subject to the same disorders as the other types of governers.

G-6 AND H-6 BRAKE VALVES

DISORDERS

48. Separation of Pressures in Brake Valve.—Fig. 8 shows sectional views of the G-6 brake valve, and Fig. 9 shows a sectional view of the H-6 brake valve. When the brake system is charged the space above the rotary valve contains air at main-reservoir pressure, the space below the equalizing piston contains air at brake-pipe pressure, and chamber *D* above the piston is charged to equalizing-reservoir pressure. Under certain conditions it is necessary to connect all these pressures, and under other conditions it is essential, in order to obtain proper brake operation, that all the pressures be separated.

The pressures in the brake valve are either permanently separated by gaskets, or are connected or separated by the movement of the rotary valve during brake operation. Gasket *32* in the G-6 brake valve and gasket *18* in the H-6 brake valve are designed to prevent leakage of air from the main reservoir or

from the brake pipe to chamber *D* or to the atmosphere at all times, and gasket *31* in the G-6 brake valve prevents a leak of

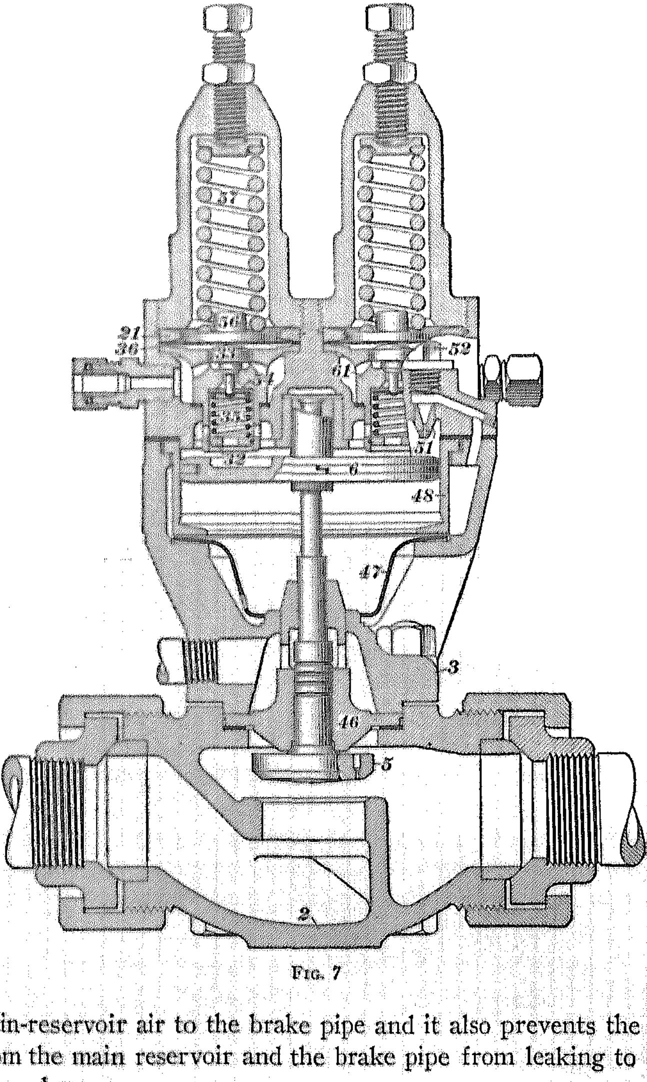
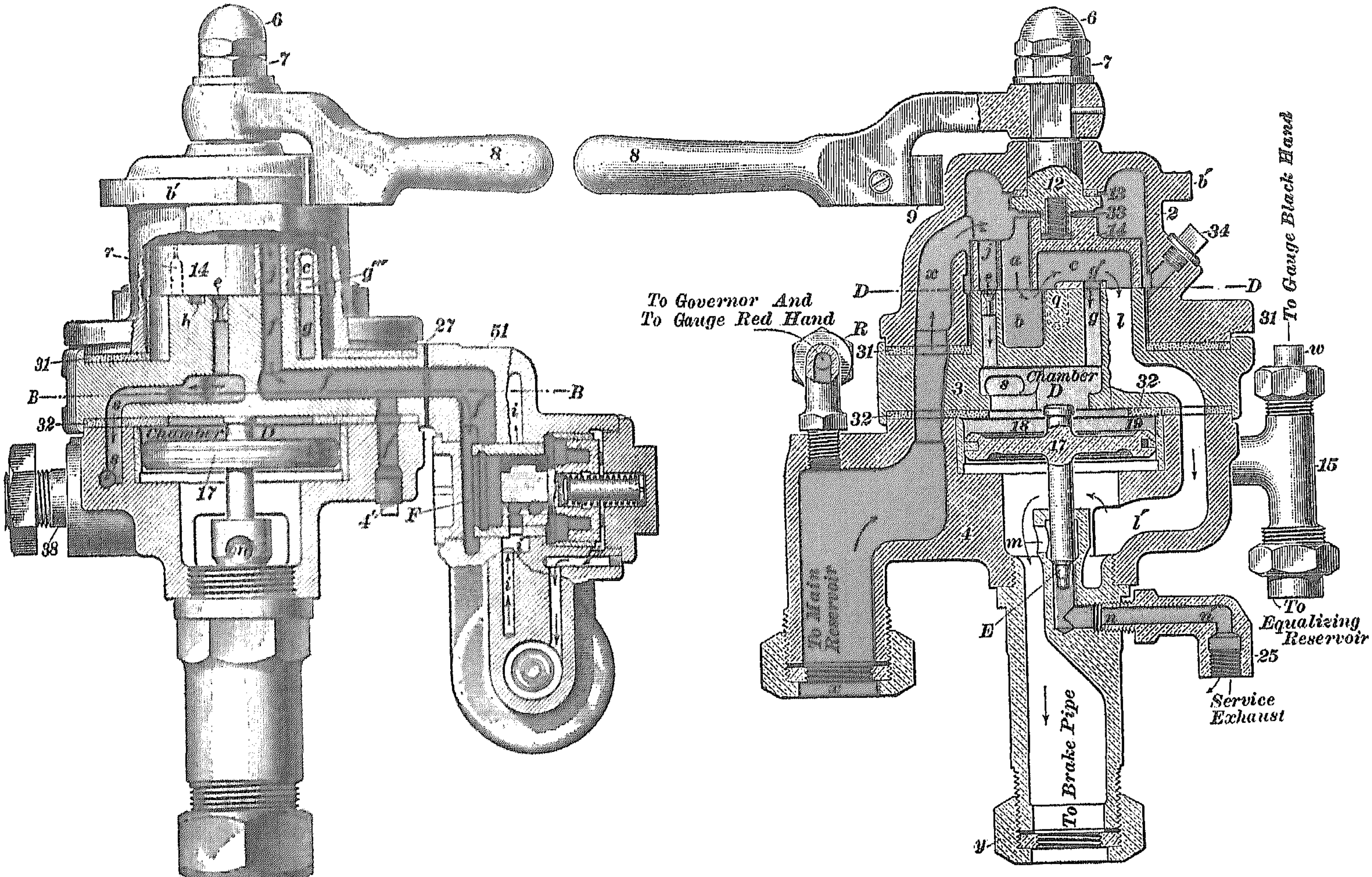


Fig. 7

main-reservoir air to the brake pipe and it also prevents the air from the main reservoir and the brake pipe from leaking to the atmosphere.


To Governor And
To Gauge Red Hand
To Gauge Black Hand
Chamber D
To Main Reservoir
To Brake Pipe
To Equalizing Reservoir
Service Exhaust

(a)

(b)

FIG. 8

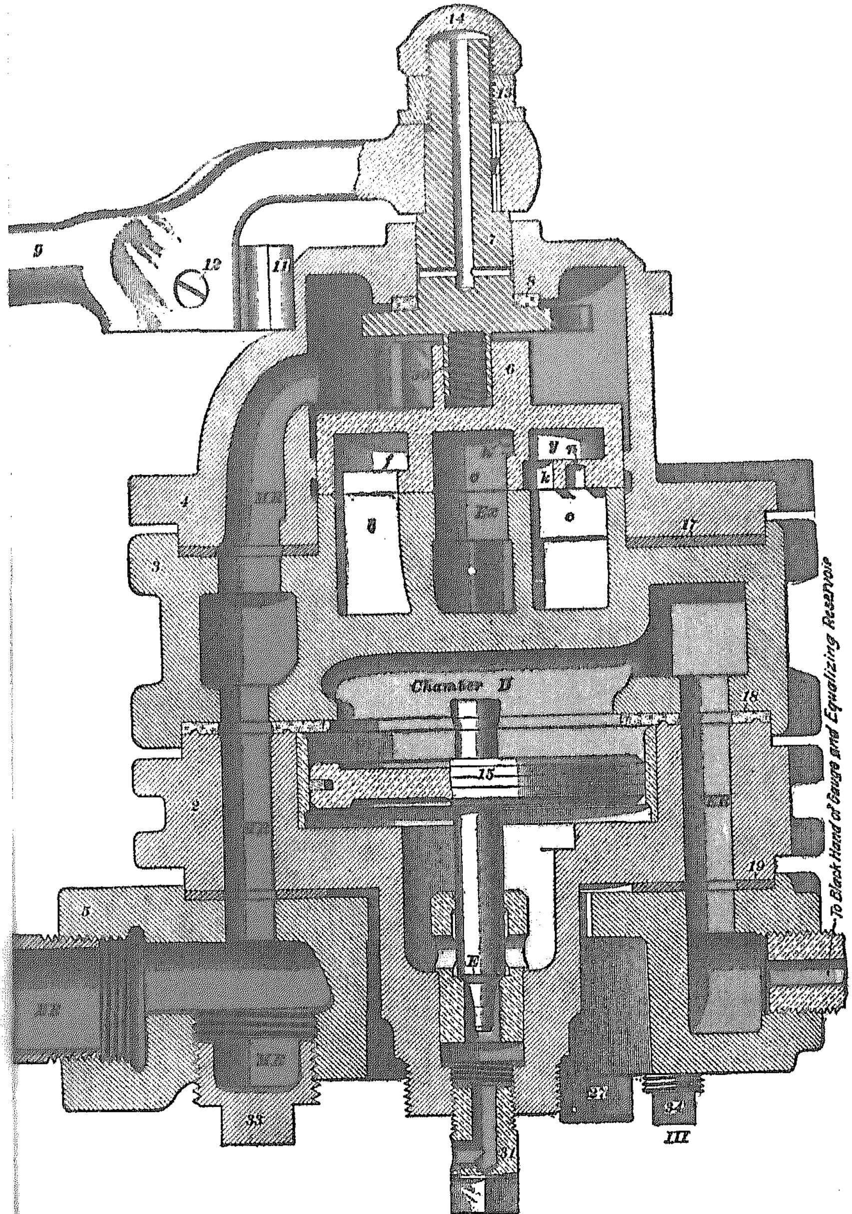


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Fig. 9

The packing ring on the equalizing piston serves at times to separate the brake-pipe pressure and the equalizing-reservoir pressure, and at times these pressures are separated by the equalizing piston when it seats against the gasket that is above it. The rotary valve separates the main-reservoir, the brake-pipe, and the equalizing-reservoir pressures, in lap, service, and emergency positions. A failure of the gaskets, or of the equalizing-piston packing ring to separate the different pressures in the brake valve, or a failure of the rotary valve to do so when required, will affect proper brake operation to an extent that will depend upon the amount of the leak. The disorders which are common to both types of brake valves will now be considered.

49. Leaky Rotary Valve.—A leaky rotary valve is one which permits main-reservoir air to leak to the brake pipe, to chamber *D*, or to the atmosphere. The brake pipe and chamber *D* are connected in running position, and a leak to one of these places will increase the pressure in both. The leak will be indicated in running position, with a short brake pipe having little leakage as with the engine alone, by the brake-pipe gauge hand registering a pressure higher than standard. With a train, brake-pipe leaks are generally such as to prevent the leak through the rotary valve from causing a noticeable increase of pressure.

The only position of the brake valve that is seriously affected by a leaky rotary valve is lap. In this position the leak tends to increase the brake-pipe pressure and thus release the brakes, especially if the train is short and the brake-pipe leakage therefore slight. The brakes on the engine and train will release with the G-6 brake valve, but with the H-6 valve only the train brakes will release. With trains of considerable length, the rotary valve rarely leaks enough to release the brakes, on account of the leak past the rotary valve being more than offset by brake-pipe leaks.

A leak to the atmosphere at the rotary valve is indicated by a blow of air at the direct-exhaust port located on the side of the brake valve.

50. Testing the Rotary Valve.—To test the rotary valve, the double-heading cock should be closed, after which the air should be exhausted from the brake pipe and the equalizing reservoir by going to emergency position. Then place the brake valve in lap position. A leak of main-reservoir air between the rotary valve and its seat to chamber *D* will be indicated by the equalizing reservoir gauge hand rising. If the leak is into the brake pipe, the increase in brake-pipe pressure will unseat the equalizing piston and cause a blow at the service-exhaust port with either brake valve.

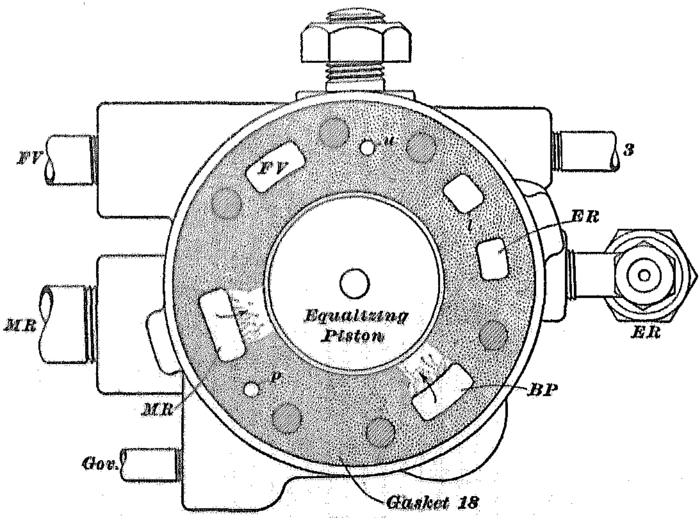


FIG. 10

A test which is less severe and which will show more accurately the effect of a leaky rotary valve under service conditions, is to leave the double-heading cock open, make a 10-pound reduction, and lap the brake valve. If the rotary valve leaks it will be indicated as already described. As some air gauges are not very sensitive, it is well to keep tapping the gauge lightly during the test, so that the leak will be more readily indicated. The brake-pipe leakage on the engine should be reduced as much as possible before making the test, thereby insuring greater accuracy.

51. Leak of Main-Reservoir Air Into Chamber D Through Gasket 32 or 18.—Gasket *18* is shown applied to the top of the bottom case of the H-6 brake valve in Fig. 10. The leak of main-reservoir air from port *MR* and brake-pipe air from port *BP* to chamber *D* above the equalizing piston is shown by arrows. Gasket *32* is shown applied to the G-6 brake valve in Fig. 11. The leak through the gasket from the main reservoir at port x' and from the brake pipe at port l is indicated in the same way as in Fig. 10. Under ordinary service conditions

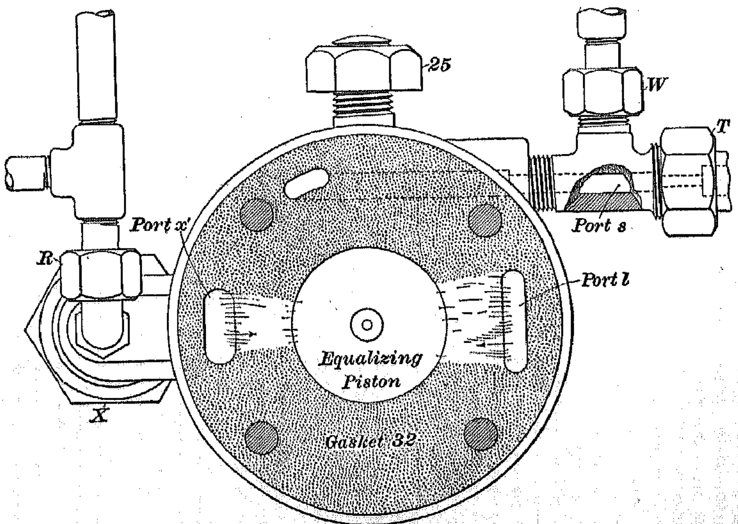


FIG. 11

gaskets *32* or *18* rarely leak, as the ribs cast on the face of the brake-valve sections on which the gaskets rest, compress the gaskets and prevent leakage. If leakage does occur it can usually be stopped by tightening the nuts on the bolts that hold the sections of the brake valve together.

The brake-valve gaskets are sometimes rendered defective by a failure to close the valve on the air-supply pipe to the power-reverse gear when using steam to operate the gear. The steam then enters the brake valve and destroys the gaskets.

A leak of main-reservoir air through gasket *32* in the G-6 brake valve, or gasket *18* in the H-6 brake valve into cham-

ber *D*, will cause the equalizing reservoir gauge hand to indicate a pressure that is higher than standard. A leak through the gaskets is most serious in service position, as the leak tends to keep chamber *D* supplied with air and the result is that the pressure in this chamber is reduced more slowly than its normal rate, which is 20 pounds in 6 or 7 seconds with an initial pressure of 70 pounds.

If the leak equals or exceeds the capacity of the preliminary-exhaust port, the equalizing piston will not rise. If the leak is less than this, the piston will seat prematurely after the brake valve is placed on lap, on account of the leak which increases the pressure in chamber *D*. The leak does not affect emergency position, as the discharge of brake-pipe air is not then dependent on a reduction in chamber-*D* pressure. Therefore, the brakes can be applied in emergency if they fail to apply in service.

It will be noted that a leak through the rotary valve to chamber *D* produces the same effect as a leak in gaskets *32* or *18*. It is of little importance that the source of the leak be determined by test, because in either case the brake valve has to be taken apart for repairs, when an inspection will quickly and positively determine which is defective. However, if the brake valve is moved from running to lap position a leak through the gaskets will generally cause the equalizing-reservoir gauge hand to rise quite rapidly.

52. Leak From Brake Pipe to Chamber D Through Gasket 32 or 18.—A leak in gasket *32*, Fig. 11, from the brake pipe to chamber *D* with the G-6 brake valve, or a similar leak in gasket *18*, Fig. 10, with the H-6 brake valve, affects the operation of the brake valve in the same way as a leak by the equalizing-piston packing ring. The effect of a leak by the ring is explained in Art. 54.

53. Leaky Equalizing-Piston Packing Ring, or Equalizing Piston Not Seating Air-Tight on Gasket.—A leaky packing ring *19* or *15* on the equalizing piston of either one of the brake valves, Figs. 8 and 9, permits air from the brake pipe to leak upwards into chamber *D* when the brakes are applied in service

on short trains. It rarely happens that the ring leaks enough to prevent the piston from lifting. If the ring leaks and if the piston fails to seat air-tight against gasket *32* or *18* when it moves all the way up, the air will leak upwards by the piston into chamber *D* when the brakes are applied on long trains. Therefore, the ring alone serves separate the brake-pipe and chamber-*D* pressures *on short trains* and these pressures are separated by the ring and by the piston seating on the gasket *with long trains.*

The reason why the pressures are separated in the above manner is as follows: When the train is short, or less than seven freight cars long, the air escapes from the brake pipe as fast as it escapes from the equalizing reservoir, and the equalizing piston seats as soon as the brake valve is lapped. In this case the equalizing piston does not lift fully and the packing ring alone serves to prevent the air from the brake pipe from leaking upwards into chamber *D*.

However, with longer trains, the air cannot escape from the brake pipe as fast as it escapes from chamber *D*. This is shown by the discharge of brake-pipe air for some time after the brake-valve handle is moved to lap. The equalizing piston then rises the limit of its travel and the flange, or sharp ridge, on its upper face beds into the leather gasket *32* or *18* that projects above it. If the gasket is in good condition an air-tight joint will be made no matter if the packing ring is worn. However, if the gasket becomes hard or cracked, the piston joint on the gasket will not be air-tight and the air from the brake pipe will leak into chamber *D*. The leak will be indicated by the equalizing reservoir gauge hand rising after the brake-valve handle is placed in lap position.

54. Effect of the Leak.—A leak at the ring and at the piston joint causes a smaller service brake-pipe reduction than is intended. For example, a reduction of 10 pounds in chamber-*D* pressure may cause a reduction of but 7 or 8 pounds in brake-pipe pressure. If the equalizing-reservoir hand of the air gauge is watched after the brake valve is lapped, a gradual increase of pressure will show due to the leak of air from the

brake pipe into the equalizing reservoir. The increase in chamber-*D* and equalizing-reservoir pressure causes the piston to seat prematurely and also lessens the brake-pipe reduction by the same amount that the pressure in chamber *D* has been increased by the leak. The piston may seat so suddenly as to release the head-end brakes on account of the increase in the pressure in the front of the train that is brought about when the flow of air from the rear of the train is stopped by the seating of the piston. As the brake-pipe pressure reduces more slowly on a long than on a short train, it follows that the leak into chamber *D* will increase with the length of the train.

55. A leak by the packing ring also has another effect: it may prevent the piston from seating the discharge valve. This disorder is indicated by a low discharge of air, which continues at the service exhaust until the brake valve is jarred. The reason that a leaky ring prevents the discharge valve from seating is as follows: When the piston moves downwards, the packing ring alone is depended on to prevent leakage from chamber *D* to the brake pipe. The difference between the pressure in these places is now very slight and if the leak by the ring is excessive and the piston is dry and dirty, the inequality of pressure on the piston will be destroyed and the discharge valve will not seat.

The increase in the equalizing-reservoir pressure which is caused by a leak of air by the piston, must not be confused with the increase in pressure which follows a continuous reduction of 20 pounds with a long train, as the increase at this time is not caused by any disorder in the brake valve. The reason for this increase of pressure, which amounts to 2 or 3 pounds, is due to the air in the equalizing reservoir cooling as it expands, thereby leaving this air cooler than the atmosphere. As it takes some time for the brake-pipe air to reduce on a long train, the air in the equalizing reservoir will have time to regain partly atmospheric temperature by absorbing heat, and causing the pressure to rise.

56. Testing Equalizing-Piston Packing Ring.—To test the equalizing-piston packing ring, a service reduction should be made with a light engine. If the discharge of air at the service-

exhaust port stops and starts at intervals when making the reduction, the packing ring is in good condition. However, if the blow is continuous until it finally stops, the packing ring is leaking. The reason why the blow is intermittent in one case and continuous in the other, is as follows: An intermittent discharge of air occurs when the ring is in good condition, because the brake-pipe air, on account of its small volume, discharges more rapidly than the air from chamber *D*, and the piston therefore keeps seating and unseating. The blow will be continuous when the brake-pipe air leaks by the ring, because the required difference in pressure is not formed on each side of the piston to seat it at short intervals.

57. Leak From Chamber D.—A leak of air from chamber *D* or from the equalizing reservoir to the atmosphere, will not be noticed in running position, because the leak will be supplied in this position. The leak is indicated by a blow of air that continues at the service-exhaust port when the brake valve is lapped after a service brake application. With the brake valve in lap position, the ports connecting the equalizing reservoir and chamber *D* to the atmosphere are closed; a leak from the equalizing reservoir or chamber *D* has the same effect as if the ports were still open. The blow at the service exhaust is caused by the leak which continues to reduce the pressure above the equalizing piston after the brake valve is lapped, and thereby causes the brake-pipe pressure to keep the piston unseated.

As the piston will not seat until all of the brake-pipe air has escaped, the double-heading cock should be closed slowly so as not to cause the head-end brakes to be kicked off and thus break the train in two by the slack running out. The trouble should be located and, if possible, remedied after the train stops.

58. Testing for Leak From Chamber D.—To test for a leak from the equalizing reservoir, make a service reduction of about 5 pounds, lap the brake valve, and note whether the discharge at the exhaust fitting stops at the proper time. It is best to make the test with a light engine. The leak may occur in the equalizing reservoir, in the pipe between the equalizing reservoir and the brake valve, in the pipe which connects the reservoir

to the gauge, including the tube inside of the gauge, or the leak may be through gasket *32* or *18* to the atmosphere. The best way to locate the leaks is to place soapsuds on the pipe joints; all the brake-valve body bolts should also be tight.

59. Broken Equalizing-Reservoir Pipe. — A brake-pipe pressure of only about 30 pounds will be obtained when the equalizing-reservoir pipe is broken between the reservoir and the T fitting at the brake valve. The reason is that the choke in the fitting permits this amount of pressure in chamber *D*.

If the equalizing-reservoir pipe is broken, or if the leak from the reservoir cannot be stopped, it will be necessary to cut the reservoir out of service. This may be done by placing a blind gasket in the equalizing-reservoir union nearest the brake valve. Before the brakes can be operated, it is also necessary to remove the service-exhaust fitting. The inner end of the fitting should be plugged and replaced.

Service position of the brake valve will not now apply the brakes, but they can be applied in service by moving the brake-valve handle slowly toward emergency position, which will connect the brake pipe directly to the atmosphere. The opening made should be sufficient to reduce the brake-pipe pressure at about the same rate as service position. To stop the discharge of air, move the brake valve slowly to service position. The reason why the port in the service-exhaust fitting is plugged is to prevent the escape of brake-pipe air when the equalizing piston is forced upwards by the brake-pipe pressure, because all the air in chamber *D* is exhausted when the brake-valve handle is moved toward emergency position. Therefore, if the port were not plugged all the air in the brake pipe would escape through the service-exhaust port as soon as the air exhausts from chamber *D*.

60. Water in Equalizing Reservoir or Nipple Almost Stopped Up.—The presence of water in the equalizing reservoir is indicated by a too rapid reduction in equalizing-reservoir pressure as shown by the gauge hand when a service reduction is made. The rapidity of the reduction will depend on how much the volume of the reservoir has been reduced by

water. This disorder is liable to cause undesired quick action when two or three car brakes are operated.

If the locomotive piping is arranged to cool the air properly before it goes to the brake valve and if the main reservoirs are drained frequently, no water will collect in the equalizing reservoir. If the nipple which is screwed into the equalizing reservoir is almost stopped up the effect will be somewhat the same as when the reservoir is partly filled with water. The black hand will fall rapidly when the brake valve is placed in service position, but in lap position the hand will go up on account of the air which keeps flowing from the reservoir to chamber *D*.

61. Blow at Service Exhaust.—A constant blow at the brake-pipe exhaust port when the brake valve is in running position is caused by dirt which holds the equalizing-discharge valve from its seat. The dirt can generally be blown off by closing the double-heading cock and then moving the brake valve from emergency position to release position a few times. The blow can also be stopped if a reduction of about 15 pounds is made and the brake-valve handle is moved slowly from lap position a short distance toward holding position with the H-6 brake valve, or from lap toward running position with the G-6 brake valve. This induces a rapid seating-and-unseating, or fluttering, action of the equalizing piston, and causes the discharge valve to seat air-tight. It may require several attempts, before the proper port opening is obtained, to produce this action. The rapidity of the fluttering action will be increased by closing the double-heading cock.

62. Service Exhaust Delayed.—The delay in the discharge of air at the service-exhaust port, when the brake valve is placed in service position, may be due to causes other than disorders in the brake valve. The discharge of air may be delayed on account of the difference in the pressure which exists at the front and rear of the train. With trains of eighty to one hundred cars there may be a difference in pressure of from 4 to 15 pounds between the two ends of the train. The difference in pressure is not dependent on the length of the brake pipe, but on the extent of the brake-pipe leaks, and where they

occur in the train. A cross-compound compressor will supply quite heavy brake-pipe leaks and will maintain about standard brake-pipe pressure at the engine, yet if the leaks were largely from the rear portion of the train, the brake-pipe pressure there would be maintained several pounds lower than on the engine.

The following explains why the difference of pressure affects the discharge of brake-pipe air: When the brake-valve handle is moved from running to service position, the passage of air to the brake pipe is prevented and the air in the head portion of the train expands to the rear where the pressure is lower, and thereby causes a reduction in pressure below the equalizing piston as fast as or faster than the pressure in chamber *D* is being reduced. The discharge of brake-pipe air at the service exhaust is then delayed on account of the time required to reduce the pressure in chamber *D* below the pressure in the brake pipe.

Although there is no exhaust from the brake pipe when the brake valve is first moved to service position, the brakes on the head portion of the train will apply on account of the fall in pressure that is produced by the air when it expands back to the rear of the train.

As the preliminary exhaust port is $\frac{1}{64}$ inch smaller in the H-6 brake valve than in the G-6 valve, the delay in the exhaust of brake-pipe air will be more noticeable with the former type of valve.

63. **Loose Handle.**—If the brake-valve handle has much lost motion on the rotary-valve key, the rotary valve will not register correctly in the different brake-valve positions. The brake-valve handle may have to be moved beyond service position before the preliminary-exhaust port will open fully, and in running position of the H-6 brake, the rotary valve may not register with the port leading to the release pipe.

OILING THE ROTARY VALVE

64. To oil the rotary valve of the H-6 brake valve, the double-heading cock should be closed in order to save the brake-pipe air. The brake valve should be placed in full-release

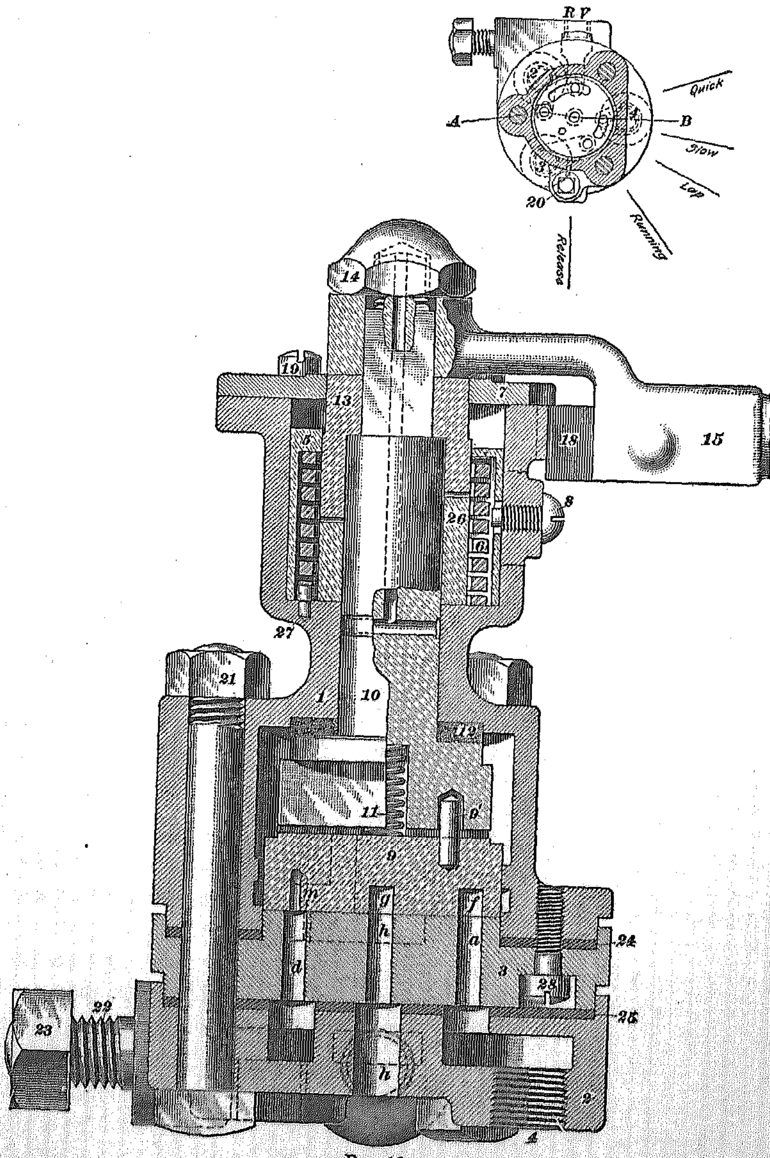

R V
Quick
A
B
Slow
Lap
20
Running
Release
14
19
13
7
18
15
5
8
26
6
27
21
1
10
12
11
9
m
g
f
h
a
d
3
24
28
22
23
25
h
2
4

Fig. 12

position and the main-reservoir cut-out cock closed, which will then bleed all the air from the brake valve. In the absence of a main-reservoir cut-out cock, the compressor should be stopped and all air drained from the main reservoir.

The oil plug in the lower side of the top case is then removed and the oil hole filled with valve oil. The brake-valve handle should be worked back and forth full stroke several times, after

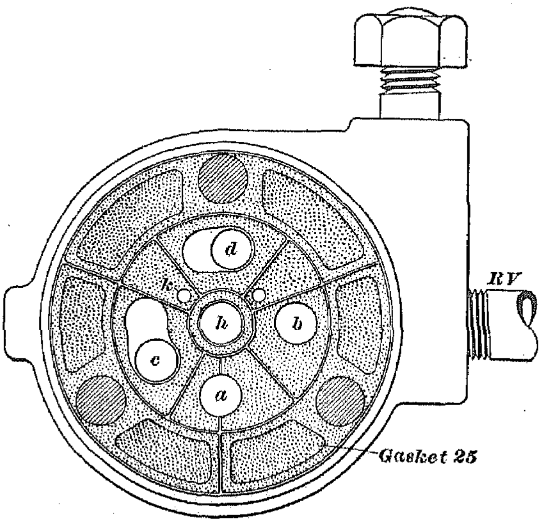


FIG. 13

which the plug should be replaced. The brake valve should be placed in lap position before the main-reservoir cut-out cock is opened. Then open the double-heading cock and place the brake valve in running position. If the car brakes are coupled, the brake-pipe pressure should be reduced 10 pounds or more below standard after the double-heading cock is opened, and the brakes then released. This is to insure the release of the brakes which will have leaked on while the work was being done. The key gasket can be oiled when there is pressure in the brake valve, by filling the oil hole in the key when the cap nut is removed. When the rotary valve of the G-6 brake valve is to be oiled, remove the oil plug *34,* Fig. 8.

S-6 INDEPENDENT BRAKE VALVE

65. Testing the Rotary Valve.—A sectional view of the S-6 independent brake valve is shown in Fig. 12. The reducing valve should be set for the required pressure and should be operating properly before the rotary valve is tested.

To make the test the brake is applied lightly and the brake valve placed in lap position. If the rotary valve leaks, the brake-cylinder hand on the small air gauge will show an increase of pressure. A leak through gasket *25*, Fig. 13, between the reducing-valve pipe *b* and the application-cylinder pipe *d*, will produce the same effect. An inspection is necessary to determine whether the rotary valve or the gasket is at fault. A blow at the exhaust port in lap position as well as in the other positions may be caused by a leaky rotary valve or a leak in gasket *25* from port *b* to port *h*.

66. Oiling the Rotary Valve.—The rotary valve is oiled by removing the oil plug *20*, Fig. 12. (See small view.) The air pressure must be exhausted from the brake valve before the oil plug is removed. The key gasket *12* can be oiled when the cap nut *14* is removed.

THE STRAIGHT-AIR BRAKE VALVE

DISORDERS

67. Blow at Brake-Valve Exhaust.—A blow at the exhaust port *Z* of the straight-air brake valve, Fig. 14, may be due to a leaky application valve *8*, a leaky release valve *9*, or a leaky washer *6*. A leak at the application valve will cause a steady blow at the exhaust port with the brake-valve handle in release position, and this leak will also cause the brakes to creep on when the handle is in lap position.

A leak at the release valve will cause a blow at the exhaust port only when the straight air is applied; a leaky washer will cause a constant blow. A blow at the exhaust port *Z* of the S-3-A straight-air brake valve, Fig. 15, when the brake is applied by the automatic brake valve, is an indication that valve *23* leaks.

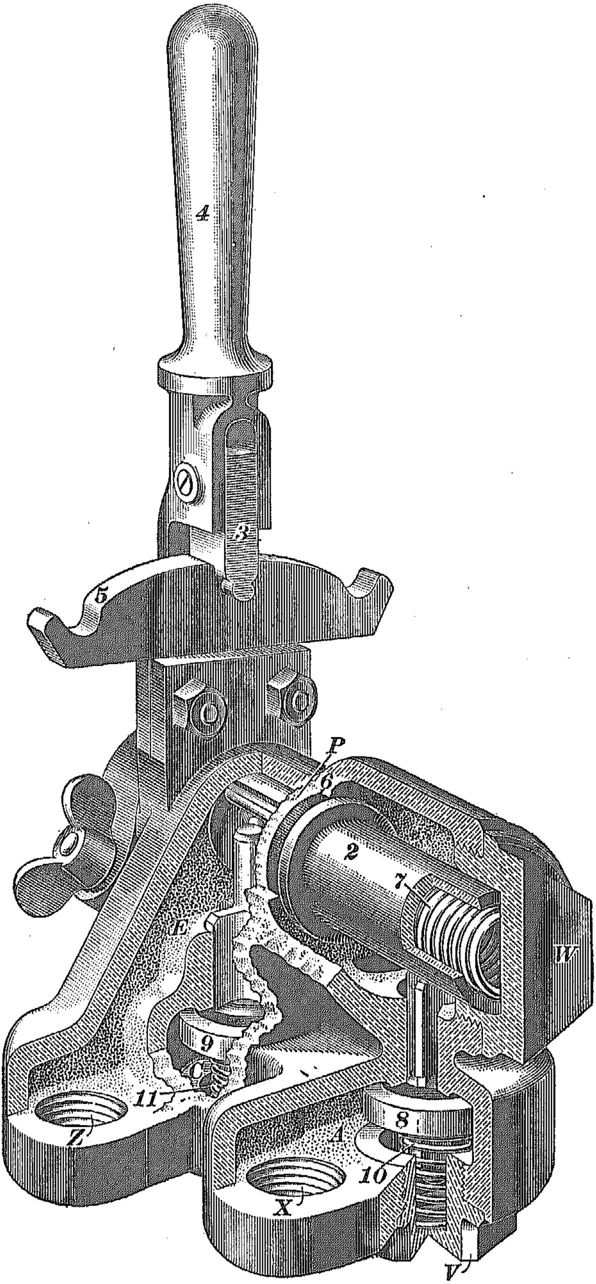


Fig. 14

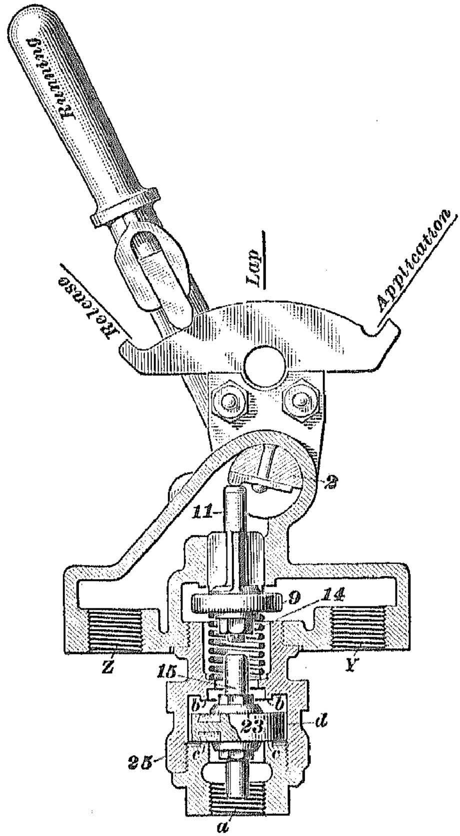

Running
Release
Lap
Application
2
11
9
14
Z
Y
15
b'
b
23
d
25
c'
c
a

FIG. 15

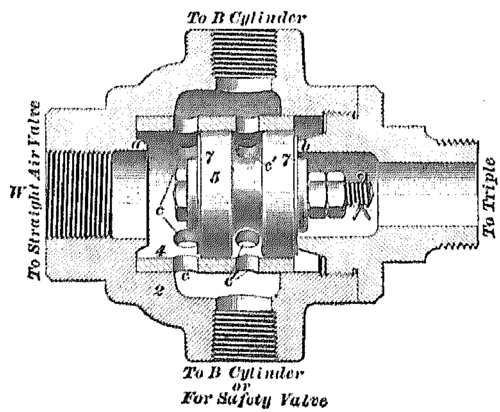

To B Cylinder
To Straight Air Valve
W
a
b
7
5
c'
7
c
4
c
c'
2
To Triple
To B Cylinder
or
For Safety Valve

FIG. 16

68. **Double Check-Valve.**—If there is a blow of air at the exhaust port of the triple valve which is on the engine, only when the straight-air brake is applied, the blow indicates that the leather gasket 7, Fig. 16, of the double check-valve on the engine does not seat air-tight at *b*. If the blow is at the exhaust port of the triple valve which is on the tender, it shows that the leather gasket of this check-valve leaks at *a*. There will be a blow at the exhaust port of the straight-air brake valve when the automatic brake is applied if there is a leak at the opposite leather-gasket of either one of the double check-valves.

TESTING AIR GAUGE

69. A test to check the accuracy of the air gauges should be made with a light engine. To make the test, place the handle of the automatic brake valve in full-release position, and then compare the pressures that are shown by the two hands of the large duplex gauge, and also the pressure that is shown on the brake-pipe hand of the small duplex gauge, where there is one. All gauge hands should show the same pressure. The gauge should be reported for repairs if a difference of over 3 pounds is shown by the gauge hands.

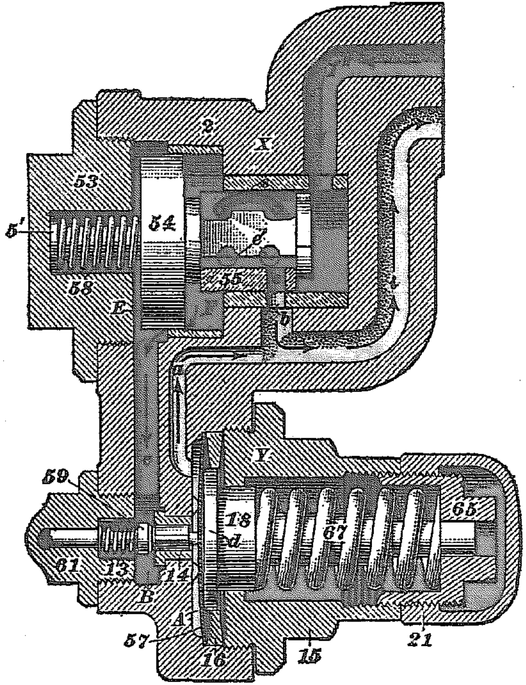
The Federal regulations prescribe that all air gauges shall be tested at least every 3 months, and whenever any irregularity is reported.

THE FEED-VALVE

DISORDERS

70. **Purpose.**—The purpose of the feed-valve is to maintain the pressure in the brake pipe at a predetermined amount when the brake valve is in running position, thereby to prevent unsupplied brake-pipe leakage from applying the brakes.

Diagrammatic views of the C-6 feed-valve are given in Figs. 17 and 18; the B-6 feed-valve is shown in Fig. 19 in closed position. A diagrammatic view of the M-3-A feed-valve is shown in Figs. 20 and 21. When studying feed-valve disorders, reference will be made to the C-6 and the B-6 valves. The M-3-A is subject to practically the same disorders as well as to a few additional ones.



BB 516B 2513

FIG. 17

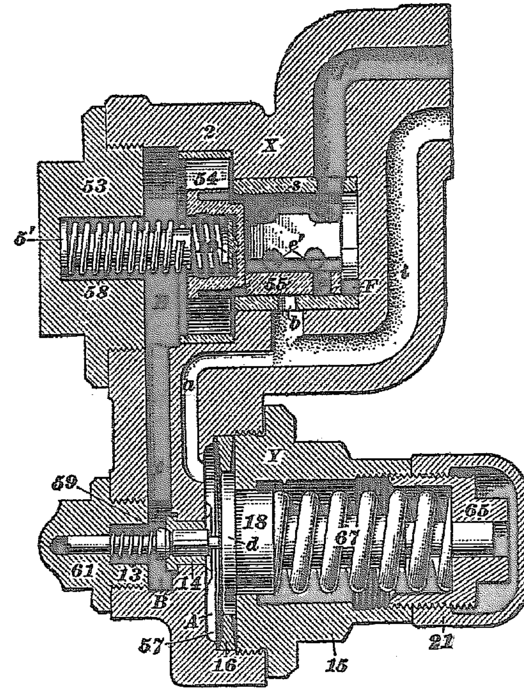


FIG. 18

The M-3-A valve was designed to overcome the tendency that the B-6 and the C-6 valve have to close at intervals before the brake-pipe pressure reaches standard.

71. Importance of Proper Maintenance.—In order to prevent an application of the brakes when it is not desired, the feed-valve should open before the leaks from the brake pipe

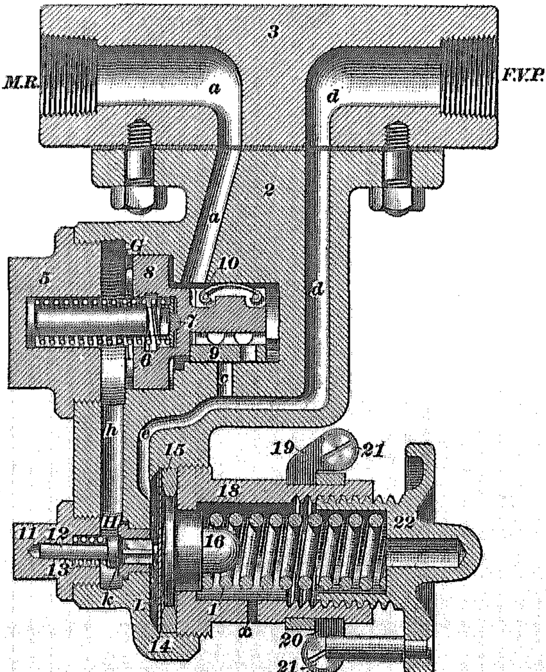


FIG. 19

reduce the pressure enough to cause the brakes to apply. It is, therefore, important to maintain the feed-valve so that it will open with a difference of pressure which will be less than that required to operate a triple valve. It should also close quickly as soon as the pressure in the brake pipe has been increased to standard. The feed-valve should open when the brake-pipe pressure is reduced less than 2 pounds, because this reduction

will cause a triple valve that is in good condition, to move to service position and to apply its brake.

72. Effects of Improperly Working Feed-Valve.—A feed-valve that works improperly will cause the brakes on the train to stick and the wheels to slide and flatten. It will also cause the brake on the engine to creep on and thus overheat the tires. A feed-valve that works improperly also has a tendency to cause undesired quick action.

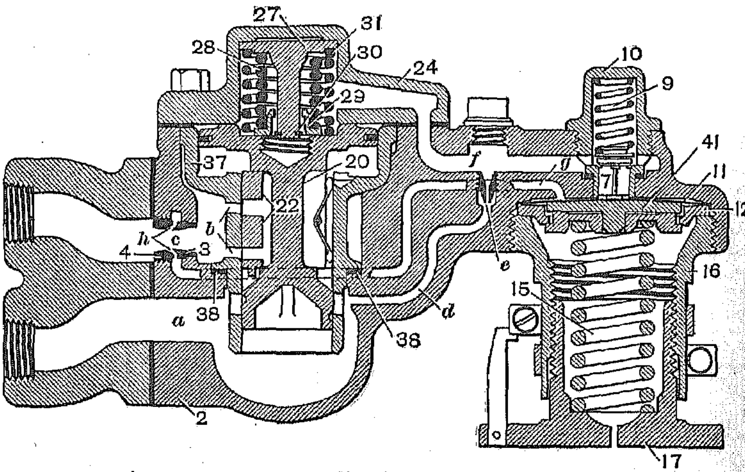


FIG. 20

73. Disorders Affecting Operation.—The disorders that affect the proper operation of the feed-valve can be traced to conditions that interfere with or render ineffective the balancing or the unbalancing of the pressure on piston *54*. This action causes the piston to fail to move the supply valve *55* at the proper time and hence results either in a pressure too high or in the pressure reducing too much before the valve opens.

The most common disorder that affects the balancing and unbalancing of the pressure on piston *54* and makes the feed-valve work improperly, is caused by the presence of gum on the piston or in its chamber, on the regulating valve *59* or on its seat, or in the air passages *a* and *c*. The tendency of gum and dirt to accumulate on these parts is increased by the close-

ness of the valve to the main reservoir. The effect of gum on feed-valve operation is aggravated by placing the valve where *it is subjected to extreme heat, which tends to harden the gum.*

74. Pressure Rises Too High.—If the brake-pipe pressure rises too high it indicates that the feed-valve has not closed soon enough. The failure of the valve to close promptly is the most common feed-valve disorder and may be due to one or a combination of the following causes: Excessive friction at the piston caused by gum or the fact that the piston fits too tight; gum on the regulating valve or its seat, which prevents the valve

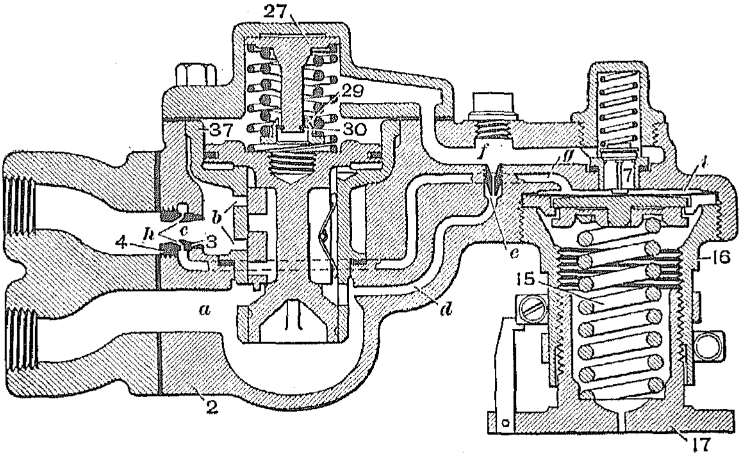


FIG. 21

from seating air-tight; a leak at either of the cap nuts *53* or *61*; a leak at the supply valve; a leak in the feed-valve gasket from the main reservoir to the brake pipe or a cracked diaphragm.

Gum increases the resistance to the movement of the piston and the supply valve, and therefore a greater difference of pressure is necessary before the piston will move the supply valve to the closed position. The brake pipe will overcharge to an extent that will depend on the difficulty with which the parts move. If the resistance of the piston is in excess of the power of the piston spring *58*, the feed-valve will remain in the open position.

When gum prevents the regulating valve from seating properly, the approximate balance of pressure necessary to move the piston to the closed position is delayed and the brake pipe will also overcharge. A leak by the regulating valve equal to or in excess of the leak by the piston will keep the supply valve open.

A leak by the piston cap *53* or regulating-valve cap *61* too small to be audible is indicated by the slight rumbling sound made by the air as it passes by the piston to supply the leak. The regulating-valve cap rarely leaks.

A leak at the supply valve or through the feed-valve gasket will not increase the brake-pipe pressure unless the leak exceeds brake-pipe leakage. When the feed-valve allows the brake pipe to overcharge, the feed-valve cannot open until the overcharge is reduced by brake-pipe leaks which, therefore, will cause the brakes to apply. When the feed-valve opens, all the brakes may not release because the brake-pipe leaks may prevent a sufficient increase in brake-pipe pressure and the result may be overheated or sliding wheels. A leak by a cracked diaphragm into the spring box in excess of the capacity of the vent port will cause the pressure to rise too high.

75. Too Much Variation Between Closing and Opening Pressure.—As already stated, the feed-valve should operate with a reduction of less than 2 pounds in brake-pipe pressure so as to prevent the brake-pipe leaks from applying the brakes. A variation in excess of 2 pounds between the closing pressure and the opening pressure may be caused by too much leakage past the piston on account of its being too loose, or to excessive friction in the regulating portion on account of the diaphragm spindle *18* not working freely through the spring, or through the regulating nut *65*. In order to reduce the resistance in the regulating portion, feed-valves are now furnished with a short diaphragm spindle *16* as shown in Fig. 19.

The piston will move slowly to open position if there is too much leakage past it, because the leak will delay the reduction of pressure behind the piston when the regulating valve opens, and it, therefore, takes longer to unbalance the pressure on the piston. The piston and supply valve will fail to move to the

open position if the leak by the piston is more than can pass by the regulating valve.

Excessive friction in the regulating portion delays the regulating valve from opening and therefore affects the piston and supply valve in the same way.

76. The reduction in the brake-pipe pressure, when the above disorders delay the opening of the feed-valve, may be sufficient for the now unsupplied brake-pipe leaks to apply the brakes, which may not all release when the feed-valve finally opens. The movement of the piston to the open position is not affected to the same extent by excessive friction as is the movement to the closed position. The reason is that the pressure reduces rapidly behind the piston when the regulating valve opens, and the piston, therefore, is forced to open position. However, the pressure behind the piston does not build up as fast when the regulating valve closes as it is reduced when the regulating valve opens. Therefore, the movement of the piston and the supply valve to the closed position will be slower than their movement to the open position.

77. Slow Feed Near Maximum Pressure.—A feed-valve that supplies the air slowly is indicated by a brake-pipe pressure which remains several pounds below standard when the pressure in the brake pipe is near the maximum pressure.

A slow feed of air may be caused by too much leakage past the piston on account of wear or unskilled repairs, or it may be due to an accumulation of gum in the air passages *a* and *c* or in the opening in which the regulating valve operates.

These disorders are indicated by a brake-pipe pressure from 4 to 7 pounds higher with a light engine than with a train that has ordinary leakage. The reason is that the small leaks on the engine can be kept fully supplied by the air that leaks by the piston and enters the brake pipe through the open regulating valve. However, the amount of air that can be supplied by a leak past the piston is too small to keep up the leaks on the train. The reason why the above disorders restrict the passage of air through the feed-valve is as follows: A leak by the piston that approaches the capacity of the regulating valve tends to

build up the pressure behind the piston and will cause it to move the supply valve toward the closed position. This restricts the passage of air through the valve and prevents the brake pipe from being fully charged to standard pressure.

When the passage of air through the air passages or past the regulating valve is obstructed by gum, the effect will be the same as if the leak were increased by the piston, and the effect on the operation of the feed-valve will be the same as if the piston were too loose.

The effect of these disorders is more apparent as the pressure nears the maximum because the quantity of air that passes the piston or the regulating valve increases with the pressure and the tendency of the piston to balance and close is, therefore, increased with the higher pressure. It will be noted that a loose piston not only causes a slow feed when the pressure is near maximum, but it also causes too much variation between the pressure at which the feed-valve closes and the pressure at which it opens. Therefore, a feed-valve that will develop one of these disorders will also develop the other.

78. Humming Feed-Valve.—The humming of the feed-valve is caused by the rapid vibrations of the brass diaphragms on account of the fact that the diaphragm ring *15* is not seating accurately against them all the way around or the diaphragms may be cracked or bulged. This disorder is usually accompanied by a blow at the hole in the spring box due to the air which leaks by the diaphragms where the diaphragm ring does not press firmly against them. The vibration of the diaphragms generally results in their being buckled. Sometimes the trouble can be remedied by merely tightening up on the spring box or by reversing the diaphragm ring or the regulating spring.

CLEANING AND OILING THE FEED-VALVE

79. To clean the feed-valve of the ET equipment with pressure in the brake system, the double-heading cock should be closed and the brake valve placed in full-release position. Close the main-reservoir cut-out cock so as to drain the air from the piping. When the brake valve and the cut-out cock are

moved in this order, the pressure is prevented from lifting the rotary valve from its seat. Remove the cap nut *53*, take out the piston and supply valve. Be careful not to bruise these parts; clean the piston and supply valve and the chambers in which they operate. The ports can be blown out by lapping the brake valve and opening the main-reservoir cut-out cock. A drop or two of valve oil should be applied to the face of the supply valve and to the spring, but no oil should be applied to the supply piston or its bushing. The cap nut should then be replaced and drawn up securely so as to be air-tight.

Remove the regulating-valve cap nut carefully, so as not to bend the regulating valve, and clean the valve and its seat. Open the cut-out cock to blow out any dirt, and replace the valve and its spring. The cap nut should be drawn up securely, but care should be taken not to put any strain on its threads. If desired, the diaphragm can be removed for examination and cleaning by taking off the spring box. It is best to slack off the regulating nut before the spring box is removed, as this makes it easier to replace the box. The diaphragms must not be bent or dented nor should they show any sign of circular cracking near the outer edge where they engage the diaphragm ring.

80. The supply valve and regulating valve can be tested by opening the main-reservoir cut-out cock. This will cause a blow at port *a* in the spring box and the blow should soon stop if the supply valve and the regulating valve are tight and there is no leak through the gasket into the brake pipe. The supply valve leaks if there is no leak at the regulating valve and the blow continues at port *a*.

The double-heading cock should be closed when cleaning the feed-valve that is used with the G-6 brake valve, and the brake valve should be placed in emergency position if there is pressure in the brake system. The brake-valve handle should be moved between running and lap positions to blow the dirt out of the valve. The repairs to the feed-valve, other than cleaning and oiling, should be made by a skilled repairman.

TESTING THE FEED-VALVE

81. The proper way to test a feed-valve is on a test rack of approved design. However, the condition of the feed-valve can be judged very accurately by making the following tests with the brake valve. It is assumed that the feed-valve has no external leakage, either at the cap nuts or in the gasket; that the air gauge is sensitive and accurate, and that the valve is adjusted for standard pressure.

82. Testing the Closing Pressure.—To test the pressure at which the feed-valve closes, make a 10-pound reduction and move the brake-valve handle to running position. Observe where the brake-pipe gauge hand stops. If the pressure rises above standard it indicates that there is excessive friction at the piston, or that there is a leak at the regulating valve.

83. Testing the Variation.—A test to determine the amount of variation that the feed-valve will allow in brake-pipe pressure before it opens, is made as follows: Make a 10-pound reduction in brake-pipe pressure, release the brake, and overcharge the brake pipe a few pounds, and return the brake valve to running position. This operation keeps the feed-valve in the closed position until leaks in the brake pipe cause it to move to the open position. The brake-pipe gauge hand will then fall a certain amount and stop, and move upwards. The pressure that the gauge registers when the hand stops is the pressure at which the feed-valve opens. This pressure is subtracted from the pressure which was obtained when testing the closing pressure. The difference between the two pressures is the feed-valve variation, or the amount that the feed-valve permits the brake-pipe pressure to reduce before it opens.

If the feed-valve is in good condition, the pressure should reduce 2 pounds, and the gauge hand should then move up rapidly. The rapid movement of the hand shows that the pressure is restored quickly. If the pressure is increased slowly, there is too much leakage at the piston or there is too much friction in the regulating portion, and the piston has not moved or has only partly moved the supply valve to open position.

With the supply valve closed, the pressure is increased slowly because the air then leaks past the piston and enters the brake pipe through the regulating valve, which is now open.

When making the above test if the recharge of the pressure chamber and the brake-pipe leaks are not sufficient to reduce the brake-pipe pressure below the point at which the feed-valve opens, it may be necessary to make a slight leak at the drain plug in the pressure chamber before beginning the test. To obtain a proper rate of leakage at an angle cock, is a rather difficult matter.

Another test for feed-valve variation, which can be easily made after a little practice, is as follows: Make a reduction of 2 pounds, move the brake valve to running position, and watch the brake-pipe gauge hand closely. If the hand moves upwards quickly it indicates that this light reduction has opened the feed-valve and that the pressure is restored quickly. If the test shows that the feed-valve does not open, further tests should be made, each with a slightly heavier reduction until the valve does open. The amount of this last reduction will show the variation in brake-pipe pressure that is necessary to open the feed-valve.

84. M-3-A Feed-Valve.—With the feed-valves just considered, the leakage by the supply piston is utilized to balance the pressure on it and cause the valve to close. The supply piston of the M-3-A feed-valve is supplied with a packing ring and the air pressure on the piston is balanced by the air that leaks through the choke *e*. Therefore, a partly or wholly stopped-up choke will, in addition to the disorders already given, cause the pressure to rise too high.

A badly worn ring on the piston will supplement the leakage by the choke and the variation between the closing and the opening pressure will be too great.

A leak by the gasket between the feed-valve and its bracket will have the same effect as a leaky slide valve. A partly stopped-up venturi tube will cause the feed-valve to keep closing and opening while the pressure is being raised to standard.

BRAKE-PIPE LEAKAGE

AIR LOSS FROM LEAKS

85. Air Gauge No Indication of Loss of Air.—The loss of air from the brake pipe by leaks is indicated by the brake-pipe hand of the air gauge, which will show a reduction of pressure. However, the reduction of pressure as shown by the gauge must not be taken as an indication of the amount of air that is lost, or of the quantity that must be supplied by the compressor to maintain the pressure. The gauges on two trains of different lengths may show an equal reduction of pressure, but it does not follow that the same amount of air is lost from each one. Therefore, the gauge alone cannot be relied upon to show the amount of air that leaks from the brake pipe.

86. Conditions on Which Loss of Air Depends.—The conditions that govern the amount of air that is lost from leakage can be more easily understood by considering two reservoirs, one of which is much larger than the other. If each reservoir is supplied with the same number of cubic feet of air per minute, the pressure will increase more slowly in the large reservoir than in the small one. The reason for this is that it requires more air to increase the pressure 1 pound in the large reservoir than in the small reservoir because the air has more room to expand in the large reservoir.

It follows then that more air must leak from the larger reservoir than from the smaller reservoir in order to reduce the pressure in each an equal amount. Therefore, the loss of air from a leak depends not alone on the extent of the leak as shown by the gauge, but also on the volume of the space from which the air escapes.

87. Rule for Loss of Air.—The following rule may be deduced from the foregoing and will apply to the loss of air from leaks: *With the same pressure and the same rate of leakage, the loss of air from the brake pipe increases or decreases accordingly as the volume of the brake pipe is increased or decreased.*

88. Application of Rule.—The following examples will show the application of this rule:

1. If trains of thirty and sixty cars are charged to the same pressure and have a leakage of 5 pounds per minute, the loss of air from the longer train will be double that from the shorter, because the volume of the brake pipe of the 60-car train is double that of the 30-car train. The compressor will have to make double the number of strokes per minute to maintain the pressure on the 60-car train.

2. The gauge shows a leak of 5 pounds per minute, and all the leakage is on the engine. If the angle cock on the tender is closed and if this reduces the volume of the brake pipe to one-tenth of the former volume, the gauge will show a leak of 50 pounds per minute. However, the compressor will have to supply the same amount of air in each case in order to keep the leak supplied.

3. If an engine had a brake-pipe leak of 5 pounds per minute and the brake-pipe volume were increased five times by coupling on cars that were tight, the gauge would show a leak of 1 pound per minute However, the same amount of air will be supplied by the compressor to maintain the pressure.

89. Calculating Loss of Air From Leakage.—The amount of free air, by which is meant its volume at atmospheric pressure that leaks from a train per minute, after the brakes are applied can be found by the following rule: *Multiply the volume of the brake pipe in cubic inches by the leak in pounds per minute. Divide by 1,728 multiplied by 15, the first number being the number of cubic inches in a cubic foot, and the latter number being atmospheric pressure. The result will be the loss of air in cubic feet per minute.*

EXAMPLE.—Find the leak of free air per minute with a train of fifty 40-foot cars and a leak of 5 pounds per minute.

SOLUTION.—On one car the brake pipe volume allowing $44\frac{1}{2}$ cu. in. for each of the two hose, and two feet for the crossover pipe is 705.8 cu. in. or 35,290 cu. in. for 50 cars. Therefore the loss of air per minute is

$$\frac{35{,}290 \times 5}{1{,}728 \times 15} = 6.7 \text{ cu. ft.} \quad \text{Ans.}$$

90. Test for Leakage.—To test for brake-pipe leakage, standard brake-pipe pressure should be obtained. A 10-pound brake-pipe reduction should be made and the brake-valve handle placed in lap position. After the brake-pipe exhaust stops, the fall of the brake-pipe gauge hand for 1 minute should be noted. This will indicate the rate of leakage. The leak should not exceed 5 pounds per minute, as this is the maximum leakage that is allowed by Federal regulations.

91. Effect of Brake-Pipe Leaks.—Leaks in the brake pipe increase the work performed by the compressor and increase the time required to charge and recharge the brake pipe. The leaks delay the release of the brakes, and assist the brake valve to apply them. After the brakes have been applied the leaks cause them to apply harder. As the brake pipe always leaks to a greater or less extent, the effect of brake-pipe leaks must always be taken into account when considering air-brake disorders, as the effect of disorders on brake operation may be increased or modified by such leaks.

OBSTRUCTION IN BRAKE PIPE

92. How Obstruction Is Indicated.—An obstruction in the brake pipe on account of a partly closed angle cock or some other cause, may be indicated in two ways; either by the discharge of air at the service-exhaust port of the brake valve becoming weak or drawn out when the brakes are applied or by the discharge of air stopping entirely and then starting again. This action is caused by the air which passes the obstruction slowly, or too slowly to keep the equalizing piston unseated. An angle cock closed on the engine or on the car next to it is indicated by a blow at the service-exhaust port when the brakes are released on account of the short brake pipe being charged faster than the equalizing reservoir.

MAIN-RESERVOIR LEAKAGE

93. **Prescribed Leakage.**—The Federal regulations prescribe that leakage from the main reservoir and related piping shall not exceed an average of 3 pounds per minute in a test of 3 minutes' duration, made after the pressure has been reduced 40 per cent. below maximum pressure. If the main reservoir is charged to a pressure of 90 pounds, this test would permit of the pressure being reduced 36 pounds, or to 54 pounds, after the compressor has been stopped.

94. **Test for Leakage.**—To test for leakage from the main reservoir, the brake valve should be placed on lap and the cut-out cock in the distributing-valve supply pipe closed, so that the brake-pipe leaks will not apply the brake and reduce the main-reservoir pressure. The red hand on the large air gauge will then indicate the extent of the leak.

MAINTENANCE OF AIR-BRAKE AND AIR-SIGNAL EQUIPMENT ON LOCOMOTIVES

Adopted as Standard, 1925

These rules were formulated jointly by the Bureau of Safety of the Interstate Commerce Commission and the Safety Appliance Committee, of the Mechanical Division of the American Railway Association. They represent minimum requirements, and shall govern the maintenance of air-brake and air-brake signal equipment on locomotives, provided that nothing herein contained shall be construed as prohibiting carriers from enforcing additional rules and instructions not inconsistent with these rules.

General Notice

The following rules shall govern the maintenance of Brake and Train Air-Signal Equipment on locomotives and cars; provided that nothing herein contained shall be construed as prohibiting carriers from enforcing additional rules and instructions not inconsistent with these rules.

Engine-House Foremen and Enginemen

1. Brake and signal equipment on locomotives and tenders must be *inspected and maintained in accordance with the orders of the* Interstate Commerce Commisssion, dated October 11, 1915; June 30, 1916; November 13, 1916; December 26, 1916; December 17, 1917, and April 7, 1919, extracts from which are prescribed in the following rules 2 to 11 inclusive (I. C. C. locomotive inspection rules 106 to 115, inclusive).

2. It must be known before each trip that the brakes on locomotive and tender are in safe and suitable condition for service; that the air compressor or compressors are in condition to provide an ample supply of air for the service in which the locomotive is put; that the devices for regulating all pressures are properly performing their functions; that the brake valves work properly in all positions; and that the water has been drained from the air-brake system.

3. *Compressors.*—(*a*) That compressors or compressors shall be tested for capacity by orifice test as often as conditions may require, but not less frequently than once each three months.

(*b*) The diameter of orifice, speed of compressors, and the air pressure to be maintained for compressors in common use are given in the following table:

Make	Size Compressor	Single Strokes Per Minute	Diameter of Orifice Inches	Air Pressure Maintained, Pounds
Westinghouse	$9\frac{1}{2}$	120	$\frac{11}{64}$	60
do.........	11	100	$\frac{3}{16}$	60
do.........	$8\frac{1}{2}$ c. c.	100	$\frac{9}{32}$	60
New York ...	2a	120	$\frac{5}{32}$	60
do.........	6a	100	$\frac{13}{64}$	60
do.........	5b	100	$\frac{15}{64}$	60

(*c*) This table shall be used for altitudes to and including 1,000 ft. For altitudes over 1,000 ft. the speed of compressor may be increased 5 single strokes per minute for each 1,000 ft. increase in altitude.

4. *Testing Main Reservoir.*—(*a*) Every main reservoir before being put into service, and at least once each 12 months thereafter, shall be subjected to hydrostatic pressure not less than 25 per cent. above the maximum allowed air pressure.

(*b*) The entire surface of the reservoir shall be hammer tested each time the locomotive is shopped for general repairs, but not less frequently than once each 18 months.

5. *Air Gauges.*—(*a*) Air gauges shall be so located that they may be conveniently read by the engineer from his usual position in the

cab. Air gauges shall be tested at least once each three months, and also when any irregularity is reported.

(*b*) Air gauges shall be compared with an accurate test gauge or dead-weight tester, and gauges found incorrect shall be repaired before they are returned to service.

6. *Time of Cleaning.*—Distributing or control valves, reducing valves, triple valves, straight-air double-check valves, and dirt collectors shall be cleaned as often as conditions require to maintain them in a safe and suitable condition for service, but not less frequently than once every six months.

7. *Stenciling Dates of Tests and Cleaning.*—(*a*) The date of testing or cleaning, and the initials of the shop or station at which the work is done, shall be legibly stenciled in a conspicuous place on the parts, or placed on a card displayed under glass in the cab of the locomotive, or stamped on metal tags. When metal tags are used, the height of letters and figures shall be not less than $\frac{3}{8}$ in., and the tags located as follows:

(*b*) One securely attached to brake pipe near automatic brake valve, which will show the date on which the distributing valve, control valve, or triple valve, reducing valves, straight-air double-check valves, dirt collectors, and brake cylinders were cleaned and cylinders lubricated.

(*c*) One securely attached to air compressor steam pipe, which will show the date on which the compressor was tested by orifice test.

(*d*) One securely attached to the return pipe near main reservoir which will show the date on which the hydrostatic test was applied to main reservoir.

8. *Piston Travel.*—(*a*) The minimum piston travel shall be sufficient to provide proper brake-shoe clearance when the brakes are released.

(*b*) The maximum piston travel when locomotive is standing shall be as follows:

	Inches
Cam type of driving-wheel brake...............	$3\frac{1}{2}$
Other forms of driving-wheel brake............	6
Engine-truck brake	8
Tender brake	9

9. *Foundation Brake Gear.*—(*a*) Foundation brake gear shall be maintained in a safe and suitable condition for service. Levers, rods, brake beams, hangers, and pins shall be of ample strength, and shall not be fouled in any way which will affect the proper operation of the brake. All pins shall be properly secured in place with cotters, split

keys, or nuts. Brake shoes must be properly applied and kept approximately in line with the tread of the wheel.

(*b*) No part of the foundation brake gear of the locomotive or tender shall be less than 2½ in. above the rails.

10. *Leakage.*—(*a*) Main reservoir leakage; leakage from main reservoir and related piping shall not exceed an average of three pounds per minute in a test of three minutes' duration, made after the pressure has been reduced 40 per cent. below maximum pressure.

(*b*) Brake pipe leakage shall not exceed five pounds per minute.

(*c*) Brake-cylinder leakage. With a full service application from maximum brake-pipe pressure, and with communication to the brake cylinders closed, the brakes on the locomotive and tender shall remain applied not less than five minutes.

11. *Train-Signal System.*—The train-signal system, when used, shall be tested and known to be in safe and suitable condition for service before each trip.

12. Enginemen when taking charge of engines must know that the brakes are in operative condition.

13. In freezing weather air compressor drain cocks must be left open while compressor is shut off.

STANDARD AIR PRESSURES

14. (*a*) Air pressure regulating devices must be adjusted for the following standard pressures:

ENGINES

(*b*)	Minimum brake-pipe pressure	70 lb.
(*c*)	Minimum differential between brake-pipe and main-reservoir pressures, with brake valve in running position	15 lb.
(*d*)	Safety valve for straight-air brake..................	45 to 55 lb.
(*e*)	Safety valve for L. T. or E. T. equipment...........	40 to 68 lb.
(*f*)	Reducing valve for independent- or straight-air brake.	40 to 50 lb.
(*g*)	Reducing valve for train air signal..................	40 to 50 lb.
(*h*)	Reducing valve for high-speed brake (minimum)......	50 lb.

CARS

(*i*)	Governor valve—water-raising system................	60 lb.
(*j*)	Reducing valve—water-raising system	20 lb.
(*k*)	Reducing valve—high-speed brake	58 to 62 lb.
(*l*)	Safety valve for L. N. and U. C. brakes.............	58 to 62 lb.

TERMINAL TRAIN BRAKE TESTS

20. Foremen of inspectors and inspectors are jointly responsible for the condition of the air-brake and train air-signal equipment on cars leaving their station.

21. The train-signal system on passenger carrying trains shall be tested and known to be in suitable condition for service.

22. (*a*) Each train must have the air brakes on all cars in effective operating condition, except in case of emergency, but at no time shall the number of operative air brakes be less than permitted by Federal requirements.

(*b*) Terminal tests of the train brake system must be made as prescribed in rules 23 to 32, inclusive, on each railroad at points where necessary to insure that the condition of the brakes is in accordance with the requirements of rule 22 (*a*).

23. Condensation must be blown from the pipe from which air is taken before connecting yard line or engine to train.

24. The train must be charged to required pressure, retaining valves, and retaining-valve pipes on freight cars inspected and known to be in suitable condition for service, and the position of angle cocks, cut-out cocks, and hose noted. A careful examination must be made for leaks and necessary repairs made to reduce leakage to a minimum.

25. (*a*) After the brake system on a freight train is charged to not less than 5 lb. below the standard pressure for that train, and on a passenger train when charged to at least 70 lb., a 15-lb. service reduction must be made upon request or proper signal, then note the number of pounds of brake-pipe leakage per minute as indicated by the brake-pipe gauge, after which the reduction must be increased to a total of 20 lb. Then an examination of the train brakes must be made to determine if brakes are applied in service application on each car; that the piston travel is correct, and that brake rigging does not bind or foul. (*b*) When the examination has been completed in accordance with rule 25 (*a*) proper release signal must be given and each brake examined to see that it releases properly.

26. Brake-pipe leakage must be reduced to the minimum, but must not exceed 7 lb. per minute.

27. Piston travel less than 7 in. or more than 9 in., must be adjusted to nominally 8 in.

28. When the test is completed the inspector or trainman who made the test will personally inform the engineman and conductor, and advise them the number of cars in train and the number having inoperative brakes.

29. Defects discovered during a standing test that cannot be repaired promptly must be reported to the foreman inspector or conductor for appropriate action in accordance with instructions of the individual carrier.

30. During standing tests brakes must not be applied or released until proper signal is given.

31. (*a*) When a train is tested from a yard test plant, an engineer's brake valve, or a suitable testing device which provides for the increase and reduction of brake-pipe pressure at the same or a slower rate as with the engineer's brake valve, should be used and be connected to the same point in the train to which the engine is to be attached.

(*b*) The train should be charged and tested as prescribed in rules 23 to 28, inclusive, and where practical should be kept charged until the road locomotive is coupled to train, when an application and release test should be made as prescribed in rule 40 for passenger trains and rule 41 for freight trains.

(*c*) If brake valve or testing device specified in rule 31 (*a*) is not used, or if after testing the brakes from a yard plant as prescribed in rule 31 (*b*) the train is not kept charged until road locomotive is coupled on, the brakes must be tested as prescribed in rule 42.

32. Before adjusting piston travel or working on brake rigging, cut-out cock in branch pipe must be closed, and reservoirs bled. Where cut-out cock is in cylinder pipe the latter only need be closed.

Road Train-Brake Tests

40. On a passenger train, before an engine is changed or an angle cock closed, except for cutting off of one or more cars from the rear of train, the brake must be applied. After recoupling and opening the angle cock and before proceeding, an application and release test must be made from the engine. Inspector or trainmen will note that the rear brakes of train apply and then signal for a release, noting that rear brakes release.

41. On a freight train, before an engine is detached or an angle cock closed on an engine or a car, the brake must be fully applied. After recoupling and opening the angle cock and before proceeding, it must be known that the brake-pipe pressure is being restored as indicated by the caboose gauge and that the rear brakes are released. In the absence of a caboose gauge, a test must be made as prescribed in rule 40.

42. At point where motive power or engine crew or train crew is changed, tests of the train brake system must be made as follows:

After the brake system on a freight train is charged to not less than 5 lb. below the standard pressure for that train, and on a passenger train to at least 70 lb., a 15-lb. service reduction must be made upon proper request or signal, brake-pipe leakage noted as indicated by the brake-pipe gauge (which must not exceed 7 lb. per minute), after which the reduction must be increased to 20 lb. Then an examination of the train brakes must be made to determine if brakes are applied in service application on each car. When this examination has been completed, proper release signal must be given and each brake examined to see that it releases properly.

43. When one or more cars are added to a train at any point subsequent to a terminal test the cars added, when in the position where they are to be hauled in the train, must be tested as prescribed in rule 42. Before proceeding, it must be known that the brake-pipe pressure is being restored as indicated by the caboose gauge and that the rear brakes are released. In the absence of a caboose gauge, a test must be made as prescribed in rule 40.

44. Before a train is operated down a grade requiring the use of retaining valves, it must be known that they are in such condition that the speed of the train can be safely controlled by the engineman.

45. Whenever the locomotive is to be detached or a stop made on a heavy grade under circumstances in which the efficiency of the air-brake system may be impaired by allowing the train to stand with the brakes applied, a sufficient number of hand brakes must be set to hold the train before the air brakes are released or the engine cut off. When ready to start, hand brakes must not be released until it is known that the air-brake system has been fully recharged.

Air-Brake Tests of Arriving Trains

46. Where inspectors are employed to make a general inspection of cars upon arrival at a terminal they must make a visual inspection of retaining valves, release valve and rods, retaining-valve pipes, brake rigging, hand brakes, hose and position of angle cocks, and make necessary repairs or mark for repair tracks any cars to which yard repairs cannot be made promptly.

47. Freight trains arriving at terminals where facilities are available and at which special instructions provide for immediate brake inspection and repairs, shall be left with air brakes fully applied. Inspection of brakes and needed repairs must be made as soon thereafter as practicable.

Double-Heading and Helper Engines

50. When more than one engine is used, brakes must be operated from the leading engine, automatic-brake valves on all except the

leading engine cut out, handles of brake valves kept in running position, and when practicable air compressors kept running.

Running Tests

51. On a passenger train, after engine or engine crew has been changed or an angle cock closed, except for cutting off cars from rear, a running test of brakes must be made as soon as speed of train permits. Such test should be made by applying the train brakes with sufficient force to ascertain whether they are operating properly. Steam or power should not be shut off unless conditions require it. In case the brakes do not operate properly in this test, the signal for brakes must be given.

FOUNDATION BRAKE RIGGING

(PART 1)

Serial 2071A Edition 1

ACTION AND CALCULATION OF BRAKE-RIGGING FORCES

LEVERS

DESCRIPTION AND CALCULATION

1. Definition.—A **lever** may be defined as any bar that is capable of being turned about a fixed point. The advantage of a lever is that it forms a convenient means of increasing or decreasing the action of an applied force. Thus, a lever may be used when required to transmit a force that is in excess of an applied force, or a lever may be employed when the applied force is too great and it is necessary to transmit a lesser force. Moving a heavy object by the use of a bar or drawing a nail from a board with a hammer are both common examples of an applied force that is increased by a single lever.

Another example of the increase of force by levers is found in the foundation brake rigging of a car. The pressure required to force the brake shoes against the wheels of a car is a great deal more than the force that can be developed by the compressed air in the brake cylinder, because the size of the brake cylinder that can be used is limited. However, by the use of a system of levers, the force that is exerted in the brake cylinder is increased to such an extent that the brake shoes are forced against the wheels with the pressure required.

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2. Action of a Lever.—The manner in which an iron bar is commonly used as a lever is shown in Fig. 1. The end of the bar is placed under the object to be moved, and a block is placed as near as possible to the same end. The object is then moved by applying force to the other end of the bar.

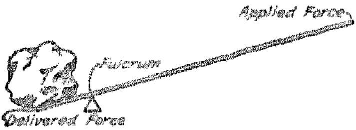

FIG. 1

The force that is applied to the lever is called the *applied force*, the force that is transmitted or delivered to the object to be moved is called the *delivered force*, and the point on which the lever turns is known as the *fulcrum*.

The point at which the force is applied and delivered, and the fulcrum point separate the lever into two parts, and these parts will be referred to as the *distance between the applied force and the fulcrum*, and the *distance between the delivered force and the fulcrum*.

The study of levers involves a consideration of the applied force, the delivered force, the force at the fulcrum, and the length of the lever between the points at which the forces act.

3. Arrangement of a Lever.—A bar can be used as a lever in three different ways, according to the relative positions of the applied force, the delivered force, and the fulcrum. The three different arrangements of levers are shown in Figs. 2, 3, and 4.

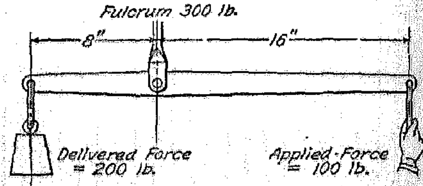

FIG. 2

In Fig. 2 the force is applied at one end of the lever, the force is delivered at the other end, and the fulcrum is between the ends.

In Fig. 3 the force is applied at one end, the fulcrum is at the other end, and the force is delivered between the ends.

In Fig. 4 the force is delivered at one end, the fulcrum is at the other end, and the force is applied between the ends.

4. Increase or Decrease of Force Dependent on Lever Arrangement.—Whether a lever will increase or decrease the force that is applied to it will depend on its arrangement or on the position of the points at which the force is applied and delivered, and the fulcrum. The force which is delivered is always more than the force which is applied when the length of the lever between the applied force and the fulcrum is greater than the length between the delivered force and the fulcrum. The more the first length exceeds the second length, the greater will be the difference between the forces. Thus, in Fig. 2, the long arm of the lever is between the applied force and the fulcrum, and an applied force of 100 pounds delivers a force of 200 pounds.

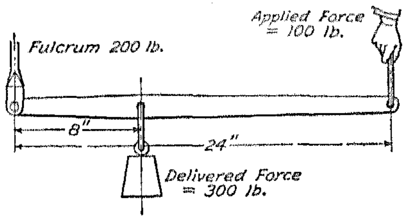

FIG. 3

With the lever arranged as in Fig. 3 the length between the applied force and the fulcrum is the whole length of the lever and cannot be less than the length between the fulcrum and the delivered force. Therefore, with this arrangement the delivered force is always more than the applied force. With this lever an applied force of 100 pounds delivers a force of 300 pounds.

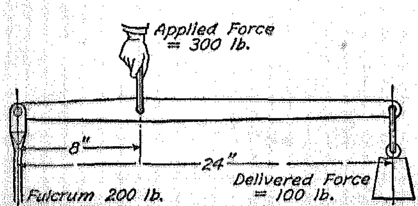

FIG. 4

When the length of the lever between the applied force and the fulcrum is less than the length between the delivered force and the fulcrum, the force delivered will be less than the force applied. This is shown in Fig. 4, in which the length of the lever between the delivered force and the fulcrum is 24 inches, and the length between the applied force and the fulcrum is 8 inches. With this arrangement an applied force of 300 pounds delivers a force of only 100 pounds.

With the lever shown in Fig. 4 the delivered force is always

less than the applied force, because the length of the lever between the fulcrum and the delivered force is the whole length of the lever and cannot be less than the length between the fulcrum and the applied force.

The lever shown in Fig. 2 can also be arranged to deliver a force which is less than the force which is applied to it. If the fulcrum point is moved until it is within 8 inches of the applied force, as in Fig. 5, the delivered force will be at the long end of the lever, and an applied force of 200 pounds will deliver a force of only 100 pounds.

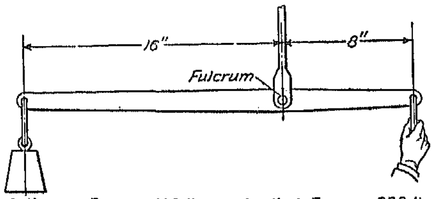

FIG. 5

The arrangement of levers shown in Figs. 4 and 5 are used in foundation brake rigging when it becomes necessary to reduce an applied force which is too great.

5. Distance Moved by Applied and Delivered Forces.—Whenever the applied force is less than the delivered force, the point at which the force is applied always moves farther than the point at which the force is delivered.

In Fig. 6 an applied force of 1 pound delivers a force of 9 pounds. When the end *a* moves from *a* to *a′*, or 1 inch, the end *b* moves from *b* to *b′*, or 9 inches. The difference in the movement of the two ends of the lever is evident, because if an applied force of 1 pound delivers a force of 9 pounds, the distance *bc* must necessarily be nine times as long as the distance *ac*.

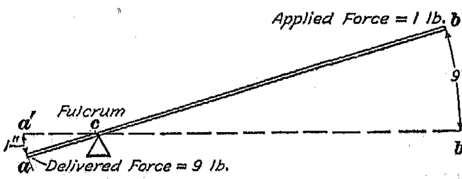

FIG. 6

In Fig. 7 if it is assumed that it is required to deliver to the car wheel a force eight times greater than the force which can be applied by the compressed air in the brake cylinder, the piston in the brake cylinder will have to move 8 inches

in order to move the brake shoe 1 inch, or the piston moves eight times as far as the brake shoe.

Therefore, when a lever delivers a force in excess of the force which is applied to it, the distance which is moved by the applied force in excess of the delivered force will depend on how much the lever multiplies the applied force.

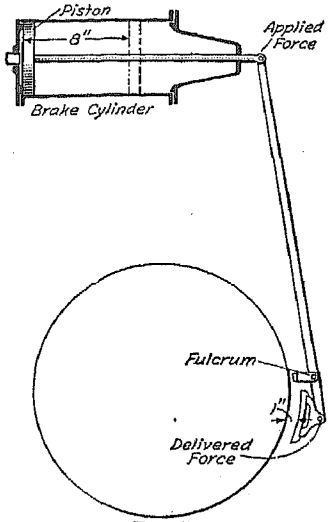

FIG. 7

CALCULATIONS INVOLVING SINGLE LEVERS

6. Forces on a Lever. The forces which act on a lever may be determined by the following rule:

Rule I.—*To obtain the force on a point between the ends of a lever, add together the forces on the ends. To obtain the force at one end, subtract the force at the other end from the force between the ends.*

Figs. 3 and 4 show that the sum of the forces on the ends of a lever is equal to the force between the ends. These figures also show that subtracting the force at one end from the force between the ends gives the force on the other end.

7. Lever Calculations.—Lever calculations are those which are necessary to determine the applied force, the delivered force, and the length of the lever between the applied force and the fulcrum, and between the delivered force and the fulcrum. It is necessary that three of these terms be given before the other one can be found. Certain problems in levers also require that the fulcrum point be located.

8. General Rule for Calculating Levers.—The general rule for calculating levers is as follows:

Rule.—*The applied force multiplied by the length of the lever between the applied force and the fulcrum is equal to the delivered force multiplied by the length of the lever between the delivered force and the fulcrum.*

From this the following rules can be deduced for calculating levers:

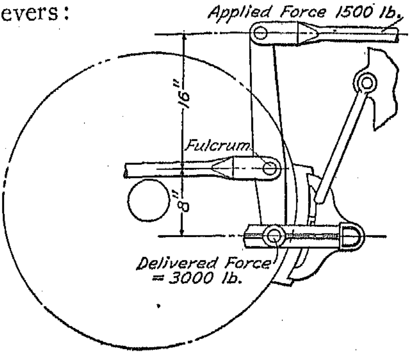

FIG. 8

Rule II.—*To find the delivered force, multiply the applied force by the length of the lever between the applied force and the fulcrum, and divide by the length of the lever between the delivered force and the fulcrum.*

EXAMPLE.—In Fig. 8 a force of 1,500 pounds is applied to the upper end of the lever; what is the delivered force with the lever dimensions as shown?

SOLUTION.—From rule II, the delivered force is equal to

$$\frac{1,500 \times 16}{8} = 3,000 \text{ lb. Ans.}$$

Rule III.—*To find the applied force, multiply the delivered force by the length of the lever between the delivered force and the fulcrum, and divide by the length of the lever between the applied force and the fulcrum.*

EXAMPLE.—In Fig. 9 the force delivered is 3,000 pounds and the lever dimensions are as shown; what force is applied to the lever?

SOLUTION.—From rule III, the force applied to the lever is equal to

$$\frac{3,000 \times 8}{24} = 1,000 \text{ lb. Ans.}$$

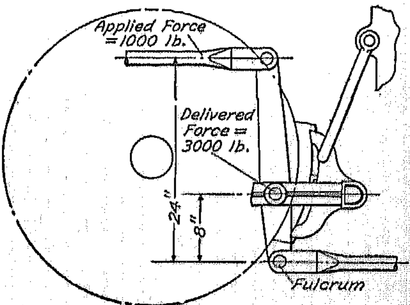

FIG. 9

Rule IV, which follows, is used when it becomes necessary

to find how far the fulcrum point is from the ends of a lever when the total length of the lever and the applied and delivered forces are known. This problem is met with only when the lever is arranged as shown in Fig. 2, because with the other two arrangements, as shown in Figs. 3 and 4, the fulcrum point is at the end of the lever.

Rule IV.—*To find the length of the lever between the fulcrum and the applied force, multiply the delivered force by the total length of the lever and divide by the sum of the applied and delivered forces.*

To find the length of the lever between the delivered force and the fulcrum, multiply the applied force by the length of the lever and divide by the sum of the applied and delivered forces.

EXAMPLE.—The total length of a lever is 30 inches. The applied force is 300 pounds and the delivered force is 700 pounds. How far is the fulcrum from the applied and the delivered forces?

SOLUTION.—From rule IV, the length of the lever between the fulcrum and the applied force is equal to

$$\frac{700 \times 30}{300+700} = 21 \text{ in. Ans.}$$

The length of the other end of the lever is then 30—21, or 9 inches. However, the length of the other end can be found by applying rule IV. Thus,

$$\frac{300 \times 30}{300+700} = 9 \text{ in. Ans.}$$

Rule V.—*To find the length of the lever between the applied force and the fulcrum, multiply the delivered force by the length of the lever between the delivered force and the fulcrum, and divide by the applied force.*

EXAMPLE.—In Fig. 10 find the length of the lever between the applied force and the fulcrum with the other lever terms as shown.

SOLUTION.—From rule V, the length of the lever between the applied force and the fulcrum is equal to

$$\frac{1{,}500 \times 24}{3{,}000} = 12 \text{ in. Ans.}$$

Rule VI.—*To find the length of the lever between the delivered force and the fulcrum, multiply the applied force by the length of the lever between the applied force and the fulcrum and divide by the delivered force.*

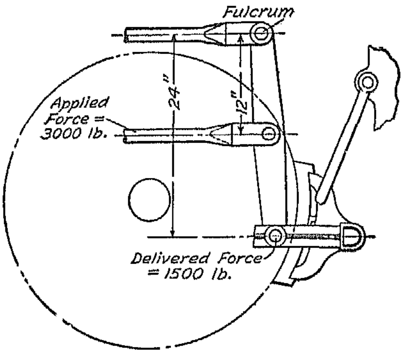

Fig. 10

Example.—How can the length of the lever between the delivered force and the fulcrum as shown in Fig. 10 be obtained?

Solution.—From rule VI, the length of the lever between the delivered force and the fulcrum is equal to

$$\frac{3,000 \times 12}{1,500} = 24 \text{ in. Ans.}$$

9. Calculating Levers Without Use of Rules.—In many cases the calculations that relate to levers may be performed mentally, and the rules do not always have to be used. By comparing the length of the lever between the applied and delivered forces and the fulcrum, the amount that the applied force is increased or decreased may be calculated quickly.

In Fig. 8 the part of the lever between the applied force and the fulcrum is twice as long as the other part; therefore, the lever must transmit 2 pounds of force for every pound of force that is applied. An applied force of 1,500 pounds, then, delivers a force of 3,000 pounds.

In Fig. 9 the length of the lever between the applied force and the fulcrum is three times as long as the part of the lever between the delivered force and the fulcrum. Each pound of applied force, therefore, delivers a force of 3 pounds, or 1,000 pounds transmits 3,000 pounds.

In Fig. 10 the length of the lever between the applied force and the fulcrum is one-half as long as the part between the delivered force and the fulcrum. Each pound of force that is applied will deliver only one-half a pound of force, or the delivered force will be one-half of the applied force.

FRICTION

DEFINITION AND THEORY

10. Definition of Friction.—A block of wood when moved over a table resists movement and a certain amount of effort is required to keep it moving after it has been started.

The resistance to movement that is encountered when one body is moved over another is called friction.

11. Theory of Friction.—The theory of friction is as follows: The surface of any material, no matter how highly it is polished, is made up of minute hills and hollows. If these elevations and depressions can be seen or felt, the surfaces are said to be *rough,* but if not the surfaces are said to be *smooth.*

When two bodies *a* and *b*, Fig. 11, are placed in contact, the inequalities on their surfaces tend to interlock, as shown. Therefore, when *a* is moved over *b* the interlocking, or meshing, action of the irregularities in the two surfaces results in a resistance to movement that is called *friction.* The greater the weight of *a* or the greater the force with which *a* is pressed against *b*, the more firmly the irregularities in the two surfaces will interlock and the greater will be the friction. The friction between two surfaces, then, depends on the condition of the surfaces and the force that is pressing them together.

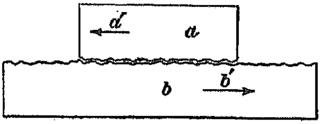

FIG. 11

It will be noted that as friction resists motion, frictional resistance will act in a direction opposite to motion when a body is moving. Thus, in Fig. 11, *a* is being moved to the left by a pull which is in the direction of the arrow *a′*. The friction between *a* and *b* causes *b* to set up a pull that acts in the direction of the arrow *b′*, and which is equal to the pull in the direction *a′*. A pull in one direction is always resisted by an equal pull in the opposite direction. This point has an important bearing on brake-shoe friction and rail friction, to be described later.

KINDS OF FRICTION

12. There are two kinds of friction: *static* and *kinetic*, and the difference between them may be more easily understood by reference to Fig. 11. If it is assumed that the pull on *a* is just enough to start this body moving from a state of rest, the friction to be overcome in starting the body is called *static friction*, or *friction of rest*. If the pull on the body is maintained, and *a* is caused to slide over *b*, the friction that is overcome is now called *kinetic friction*, or *friction of motion*.

Less effort is required to keep a body moving than to start it moving. In other words, kinetic friction is always less than static friction. The explanation is that when a body is at rest the elevations and depressions in the two surfaces have a chance to become more or less thoroughly interlocked and considerable effort is required to pull one surface free from the other. However, as soon as one body is started the elevations on one surface do not have time to fit into the depressions on the other surface, and the resistance to movement decreases.

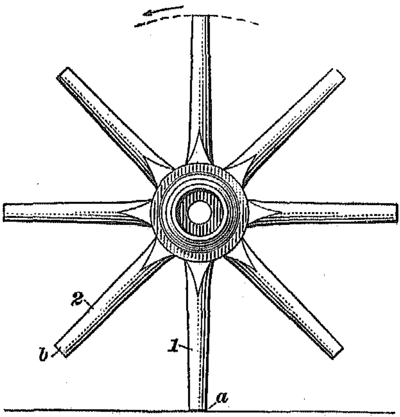

FIG. 12

13. Rolling Motion and Rolling Friction.—To understand what is meant by rolling friction, it is necessary to consider, briefly, rolling motion. Fig. 12 shows a wheel without the tire. If the wheel is turned the motion is not exactly rolling, but rather a series of rotations about the ends of the spokes as they come in contact with the ground. Thus, as shown, the entire wheel is rotating about the end *a* of the spoke *1*, but in the next instant the rotation will be about the end *b* of the spoke *2*.

For an instant, then, the end of the spoke on the ground is

not moving but is at rest, and the whole wheel turns about the point that is at rest. If the number of spokes were multiplied a great many times, the wheel would roll smoothly over the ground, but the character of the motion would remain the same; that is, for each instant there would be no movement between the ground and the end of the spokes.

When a tire is applied to the wheel, it is not difficult to imagine the tire as being made up of a great number of spokes which are placed very closely together. Then, for an instant, the part of the tire in contact with the ground is at rest with respect to the ground, and the wheel is rotating as a whole about the point in contact. The photograph of a wheel rolling on a rail shows the part of the wheel near the rail clearly and sharply defined, while the upper part of the wheel is blurred. This is an indication that the part of the wheel which is in contact with the rail is at rest relatively to the camera. Therefore, the friction between the part of a car wheel that is resting on a rail is friction of rest and not friction of motion. Instead of referring to this friction as static friction, it is called *rolling friction*, on account of the rolling motion of the body.

Rolling friction is, therefore, another name for static friction, and this term will be used with the meaning of static friction when the friction between a wheel and a rail is considered.

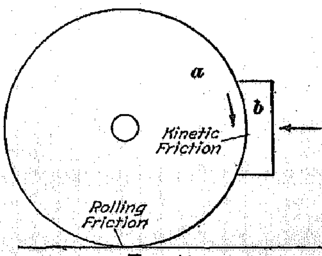

FIG. 13

14. Rolling and Kinetic Friction Acting in Opposition.—A brake shoe when it is applied to a revolving car wheel furnishes an example of rolling and kinetic friction acting in opposition. In Fig. 13 if we assume that the wheel shown is turning on the rail in the direction indicated by the arrow, the friction between the wheel and the rail is rolling friction.

The friction between the brake shoe *b* and the wheel will be considered next. When the brake is applied, the friction between these parts is kinetic friction, or the friction of motion,

because the wheel is turning against the stationary shoe. The brake shoe may be compared to *b* in Fig. 11, and the wheel to *a*. Therefore, the rolling friction between the wheel and the rail tends to keep the wheel turning and acts in opposition to the kinetic friction between the brake shoe and the wheel, which tends to stop the wheel from turning. The rolling friction between the wheel and the rail is called *adhesion*, when the maximum friction between these parts is meant.

FRICTION PRODUCED BY BRAKES

15. How Friction Stops a Train.—A *brake* is a device that depends on friction to stop or retard motion. The friction that results from the application of the brake shoes to the wheels is usually regarded as being the sole factor in stopping a train. However, it will now be shown that another factor must receive consideration.

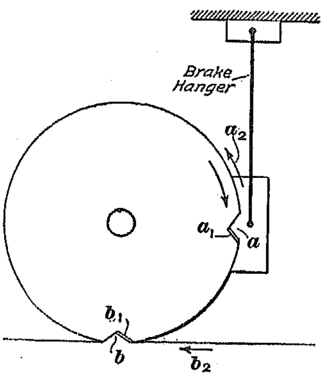

FIG. 14

It has already been mentioned in Art. **11** that friction always acts to set up an equal pull in the direction opposite to the pull that produces motion. The operation of this action on a car wheel will be explained by considering Fig. 14. This figure shows a car wheel which is assumed to be moving in the direction shown by the arrow on the wheel.

The pull that is necessary to keep the wheel turning causes the part of the wheel in contact with the rail to exert an equal backward push on the rail. In other words, the wheel pushes backward on the rail with the same force that the wheel is pulled ahead. It will now be shown how the above condition changes when the brake is applied.

It is assumed in Fig. 14 that the total friction between the brake shoe and the wheel is equal to that which is produced

by the elevation a on the shoe when it interlocks with the depression a_1 in the wheel; likewise the friction between the wheel and the rail is equal to the friction that results from the interlocking of b and b_1. When the elevation a interlocks with the depression a_1, the friction produces a pull on the wheel that tends to stop it from turning. The direction of this pull is shown by the arrow a_2. When a interlocks with a_1, the depression b_1 on the wheel is forced against the elevation b on the rail. The resulting friction causes the wheel to push forward against the rail, or the rail to push backward against the wheel with a force equal to the pull of the brake shoe on the wheel. The direction of this backward push is shown by the arrow b_2. Therefore, the retarding force that acts on a train to stop it is the friction between the wheels and the rails or the rail pull, and this friction or pull is caused by and is equal to the friction between the brake shoes and the wheels.

When trains are being stopped the friction that causes the push backward on the wheel is generally referred to as *rail pull,* and the friction between the brake shoe and the wheel as *wheel pull.* When these terms are used, the force that stops a train is the rail pull, which is equal to the wheel pull or the pull or push on the brake hangers. There would be a pull on the brake hangers when the brake shoe is hung as in Fig. 14. However, there would be a push on the hanger with the brake shoe behind the wheel, and the wheel turning in the same direction, as in Fig. 14.

16. The rail friction is in reality a more important factor than the brake-shoe friction, because the brake-shoe friction is limited by the rail friction. It would be useless to have more brake-shoe friction than rail friction, because the wheels would slide as soon as the brakes were applied.

The importance of rail friction in the stopping of a train may be more easily understood by considering the rails as being greased or made of some material that offers little friction such as ice. The brakes would apply with their usual pressure, but because of the absence of rail friction the wheels would immediately slide and very little resistance would be

offered to stop them. Therefore, if it were not for the rail friction, the friction which is developed by the brake shoes would be of no effect.

COEFFICIENT OF FRICTION

17. The meaning of the term *coefficient of friction* as applied to the air brake will be explained by considering Fig. 15. This figure shows a brake shoe connected to a spring balance by a hanger. The wheel is assumed to be turning in the direction of the arrow and it is also assumed that the brake shoe is being pressed against the wheel with a constant pressure of 1,000 pounds. The friction between the surfaces of the brake shoe and the wheel causes a downward pull on the shoe, and this pull is communicated to the spring balance through the hanger. If it is assumed that the pull registered by the spring balance is 100 pounds, the friction between the brake shoe and the wheel is 100 pounds, or 100 pounds of the 1,000 pounds brake-shoe pressure acts as friction to stop the rotation of the wheel.

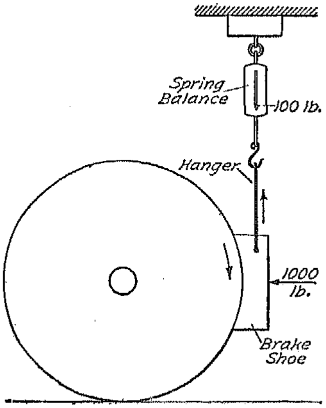

FIG. 15

Instead of saying that so much friction is obtained from some certain pressure, it is convenient to use a term that will indicate the friction that results from each 100 pounds of pressure. This makes it necessary to compare the brake-shoe friction with the brake-shoe pressure. The comparison is made by dividing the friction by the pressure. Thus $\frac{100}{1000}=\frac{1}{10}$, or 10 per cent., and the friction is, therefore, 10 per cent. of the brake-shoe pressure. The comparison between the brake-shoe friction and the brake-shoe pressure expressed as per cent. is called the coefficient of friction.

The coefficient of friction in the preceding example is 10 per cent., because a comparison between the friction and the

pressure shows that the former is 10 per cent. of the latter. A coefficient of friction of 10 per cent. then, means that each 100 pounds of brake-shoe pressure gives a brake-shoe friction of 10 pounds. In other words, a force of 10 pounds acts to stop the wheel from turning for each 100 pounds of force that is applied to the brake shoes.

It is important to know what the coefficient of friction is, because when its value is given, it is a simple matter to find the brake-shoe friction when the brake-shoe pressure is given. It is the practice in the solution of air-brake calculations to express a per cent. as a decimal, thus: 10, 15, or 25 per cent. is written as .10, .15, or .25.

18. Rule for Finding the Coefficient of Friction. To find the coefficient of brake-shoe friction, it is necessary to ascertain what per cent. of the brake-shoe pressure the brake-shoe friction is. This can be done by the use of the following rule:

Rule.—*To find the coefficient of friction, multiply the brake-shoe friction by 100, and divide by the brake-shoe pressure.*

EXAMPLE.—The brake-shoe friction, as indicated by the pull on the brake-beam hangers, is 500 pounds, and the brake-shoe pressure is 2,500 pounds. What is the coefficient of friction?

SOLUTION.—From the rule, the coefficient of friction is

$$\frac{500 \times 100}{2{,}500} = 20 \text{ per cent. Ans.}$$

19. Distinction Between Friction and Coefficient of Friction.—A distinction should be made between the terms *brake-shoe friction* and *coefficient of brake-shoe friction,* and care should be taken not to confuse their meaning. It is correct to use the term brake-shoe friction when it is not desired to show the comparison between it and brake-shoe pressure; thus, it can be said that brake-shoe friction is the result of brake-shoe pressure.

The term coefficient of friction is to be used when it is desired to compare or to show the relation between brake-shoe friction and brake-shoe pressure. Thus the statement

that the coefficient of friction is 10 per cent., means that the brake-shoe friction is $\frac{1}{10}$ or 10 per cent. of the brake-shoe pressure, or that the pressure is ten times greater than the friction.

20. Rule for Finding the Brake-Shoe Friction. The friction between the brake shoe and the wheel is equal to the pull or the push in pounds on the brake-beam hanger. When it is required to find the brake-shoe friction, it is necessary to multiply the brake-shoe pressure by the coefficient of friction. The brake-shoe friction can then be determined from the following rule:

Rule.—*To find the brake-shoe friction, multiply the brake-shoe pressure by the coefficient of friction.*

EXAMPLE.—The brake-shoe pressure is 6,000 pounds, and the coefficient of friction is 15 per cent. or .15. What is the friction that acts to stop the rotation of the wheel?

SOLUTION.—From the rule, the friction equals $6,000 \times .15 = 900$ lb. Ans.

The brake-shoe friction is always much less than the brake-shoe pressure. This is shown by the size of the brake levers, pins, and brake beams through which the brake-shoe pressure is transmitted as compared with the size of the brake-beam hangers which resist the pull which results from the brake-shoe friction.

21. Conditions Affecting the Coefficient of Friction.—The coefficient of friction is affected by and depends on: (*a*) the pressure with which the brake shoes are forced against the wheels; and (*b*) on the condition of the surfaces of the brake shoes and the wheels that engage. If the brake-shoe pressure is maintained constant, any change in the coefficient of friction must be due to a condition that affects the surfaces in contact. The application of the brake shoes to the wheels when stopping a fast, heavy train causes the shoes to become very hot. Therefore, the heating of the brake-shoe metal that is opposed to the wheels is the sole cause of the change in the coefficient of friction. The temperature of a brake shoe depends on the pressure against it, the speed of the wheel, and the time the shoe is in contact with it.

22. Variation in Coefficient of Friction During a Stop.—When a variation in the coefficient of friction is considered, it will be assumed that a stop is being made with a modern fast passenger train. The coefficient of friction varies throughout the stop, and it is very difficult to determine its exact value on account of the rapid changes in the temperature of the surfaces of the brake shoes. When the brakes are first applied the coefficient of friction increases, then it decreases and remains fairly constant, and, as the speed decreases to about 30 miles per hour, it begins to increase again and continues to do so until the train stops. The reason for this variation is as follows: When the brakes are applied the heating of the brake shoes causes the coefficient of friction to increase, and it is estimated that it reaches its maximum when the temperature of the rubbing surfaces reaches about 800° F. As the temperature goes higher, the metal in the brake shoes becomes so hot that it begins to weaken, and, under severe conditions, it may even discharge from the brake shoes in a molten condition. The coefficient of friction decreases at this time because the elevations in the surfaces that are in contact are more easily rubbed off. The coefficient of friction then remains fairly constant even with the rubbing surfaces in a molten condition because, as the metal weakens or is thrown off, a new surface is presented to the wheel, which breaks down in turn and causes another surface to be set up. As the speed reduces the slower rotation of the wheels permits the brake shoes to cool off, until the metal in the shoes is no longer thrown off or has its abrasive qualities weakened by heat. As the brake-shoe metal returns to its normal condition the coefficient of friction will keep increasing until the stop is completed.

23. When a high braking force is used to stop a train at a relatively low speed, the coefficient of friction increases from the time the brakes are applied until the train stops; the reason is that the surfaces in contact do not become so hot and this results in a better interlocking of the brake shoe and wheel metal. The fact that the coefficient of friction is greater at

low than at high speed explains why such severe shocks result from heavy brake applications and why wheels will slide at low speed and not at high speed.

It must be remembered that the coefficient of friction is a measure of the retarding action of the brake shoes. Therefore, in the foregoing when the coefficient of friction is low, the retarding action of the brakes is also reduced, and the opposite occurs when the coefficient of friction increases. The reason that the brake shoes wear out so rapidly under severe braking conditions without affecting the wheels, is on account of the harder and tougher nature of the wheel, and because the surface of the wheel is not continuously in contact with the brake shoe, while the brake shoe is in continuous contact with the wheel.

24. Mean Coefficient of Friction.—As already stated, the coefficient of friction is constantly changing while the train is being stopped, and it is very difficult to determine its value at any one time. However, tests have shown that the mean or average coefficient of friction, when a stop is made with a modern passenger train, is approximately 10 per cent. In freight service the average coefficient of friction is considerably higher and may run as high as 15 or 18 per cent.

The statement that the mean coefficient of friction is 10 per cent. emphasizes the small amount of friction or retarding force that is actually developed by the brake shoes. The statement will have more significance when it is stated that only 1 pound acts as friction for each 10 pounds of brake-shoe pressure. This means that the efficiency of the brake shoes is only 10 per cent.

ADHESION

25. Coefficient of Adhesion.—It has already been explained that the friction between the wheels and the rail causes the wheels to turn when the car is being moved. This friction is termed *rolling friction* and the term *adhesion* refers to the maximum rolling friction between the wheel and the rail.

The term *coefficient of adhesion* will be explained by considering a wheel weighing 1,500 pounds. If the maximum fric-

tion or adhesion between the wheel and the rail is 375 pounds, it would require a brake-shoe friction of this amount to slide the wheel. A comparison between 375 and 1,500 shows that the former is $\frac{1}{4}$, or 25 per cent. of the latter. Thus, $\frac{375}{1500} = \frac{1}{4}$, .25, or 25 per cent.

The term coefficient of adhesion is applied to the comparison between the adhesion between the wheel and the rail, and the weight on the rail under the wheel, expressed as per cent. In the above example the coefficient of adhesion is 25 per cent., because a comparison between the adhesion and the weight on the rail shows that the former is 25 per cent. of the latter. The importance of knowing the value of the term is that it enables one to calculate the adhesion when the weight on the rail under the wheel is given. The coefficient of adhesion depends entirely upon the condition of the rail, and if this does not change the coefficient of adhesion will remain the same. When the rails are dry and clean the coefficient of adhesion is about 25 per cent., but if the rails are wet the coefficient of adhesion may be as low as 15 per cent. An average value is 20 per cent. The coefficient of adhesion may be determined by using the following rule:

Rule.—*To find the coefficient of adhesion, multiply the adhesion by 100, and divide by the weight under the wheel.*

EXAMPLE.—The weight on the rail under a wheel is 6,000 pounds, and the adhesion between the wheel and the rail is 1,200 pounds. What is the coefficient of adhesion?

SOLUTION.—From the rule the coefficient of adhesion is equal to

$$\frac{1{,}200 \times 100}{6{,}000} = 20 \text{ per cent. Ans.}$$

26. Difference Between Adhesion and Coefficient of Adhesion.—The terms adhesion and coefficient of adhesion should not be confused. The term adhesion does not imply any comparison between the adhesion and the weight that is on the rail under the wheel, while the term coefficient of adhesion does.

27. Finding the Adhesion.—When the coefficient of adhesion and the weight of the wheel on the rail are given, it is an easy matter to calculate the adhesion or the total frictional force between the wheel and the rail. In Art. **25** the wheel was assumed to weigh 1,500 pounds and the coefficient of adhesion was 25 per cent., or .25. If it is required to find the total frictional force between the wheel and the rail, it is only necessary to multiply 1,500 by .25. The result is 375 pounds, or the total frictional resistance between the wheel and the rail. The following rule can be deduced from the foregoing:

Rule.—*To find the total frictional force between the wheels and the rail, multiply the weight under the wheels by the coefficient of adhesion.*

EXAMPLE.—A car weighs 80,000 pounds. What is the adhesion when the coefficient of adhesion is 20 per cent.?

SOLUTION.—The adhesion is equal to $80,000 \times .20 = 16,000$ lb. Ans.

28. Braking Force Limited by Adhesion.—The adhesion, or the total frictional force between the wheel and the rail, limits the braking force that can be applied to the wheel. The adhesion rarely exceeds 25 per cent. of the weight under the wheels and the friction of the brake shoes must not equal this amount, or the wheels will slide. In fact, the brake-shoe friction should be less to allow for a bad condition of the rails.

Under ordinary conditions there is little danger that the friction between the brake shoe and the wheel will exceed the friction between the wheel and the rail. Sixty per cent. of the empty weight of a freight car is used as the braking force; that is, for each 100 pounds of weight, 60 pounds is used as braking force. The coefficient of friction by which 60 pounds must be multiplied in order to give a frictional resistance of 25 pounds, is 41 per cent., or $60 \times .41 = 24.6$ per cent. This is a greater frictional resistance than the brake shoes can develop. The following example will show the difference that exists between the frictional resistance at the brake shoes and at the rail:

EXAMPLE.—The brake-shoe pressure that is applied to the wheels of a car that weighs 80,000 pounds is 72,000 pounds. Find the difference

between the total frictional resistance at the rail and at the brake shoes, if the coefficient of adhesion is 25 per cent., and the coefficient of friction is 20 per cent.

SOLUTION.—The total frictional resistance at the rail is 80,000×.25, or 20,000 lb. The total frictional resistance at the brake shoes is 72,000×.20 or 14,400 lb. The difference is 20,000−14,400=5,600 lb. Therefore, there is a margin of 5,600 pounds of rail friction, which acts to prevent the wheels from sliding.

SLIDING WHEELS

29. Reason For a Wheel Sliding.—Before a wheel will stop revolving and begin to slide, the total frictional resistance between the brake shoe and the wheel must exceed the total frictional force, or the adhesion between the wheel and the rail.

The total frictional resistance between the brake shoe and the wheel is equal to the brake-shoe pressure multipled by the coefficient of friction, and the adhesion, or the total frictional force between the wheel and the rail, is equal to the weight on the rail under the wheel multiplied by the coefficient of adhesion.

When the brake is applied, the friction between the brake shoe and the wheel which tends to stop its rotation instantly sets up an equal and opposing friction between the wheel and the rail that resists any tendency to prevent the wheel from turning. That is, any tendency to stop the wheel from turning in one direction is resisted by the push of the rail against the wheel in the direction opposite to the movement of the train. However, the wheel will slide when the total frictional resistance between the brake shoe and the wheel becomes equal to or slightly exceeds the total frictional resistance between the wheel and the rail. In other words, the wheel will slide when the wheel pull exceeds the rail pull. Fig. 16 will serve to make the foregoing clear.

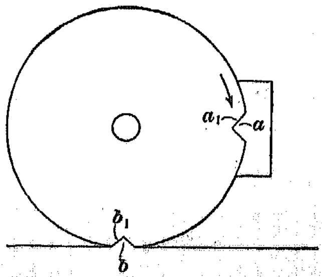

FIG. 16

It is assumed that the total frictional resistance between the brake shoe and the wheel is equal to that which is caused by the elevation a on the brake shoe when it engages the depression a_1 in the wheel. Likewise, the total frictional resistance between the wheel and the rail is assumed to be equal to that which is due to the elevation b on the rail when it engages the depression b_1 on the wheel.

The elevation a tends to stop the wheel from turning and this action of a forces the wheel against the elevation b, which resists the tendency produced by a. However, the frictional resistance exerted by a, as a is the greater, will exceed the frictional resistance that is exerted by b. The wheel, therefore, tears off the projection b and slides along the rail, because the wheel is prevented from turning by the projection a.

30. Turning and Sliding Wheels.—A wheel with the brake applied will always stop in a shorter distance when it is turning than when it is sliding. The reason is that rolling friction is static friction, and static friction is always greater than kinetic friction, or friction of motion. As long as the wheel turns, the friction between the wheel and the rail is rolling friction and the friction between the brake shoe and the wheel is kinetic friction. The instant the wheel begins to slide the rail friction becomes kinetic and the brake-shoe friction becomes static. The frictional resistance between the wheel and the rail immediately decreases and the result is that the wheel goes farther before it stops, when it is sliding than when it is turning.

BRAKE-RIGGING FORCES

BRAKING FORCE

31. Definition.—The term braking force refers to the force with which the brake shoes are forced against the wheels. Braking force and brake-shoe pressure mean the same. Braking force was formerly known as braking power, but this latter term is incorrect, as power refers to the rate of doing work,

and its unit of measurement is the horsepower, whereas the braking force or brake-shoe pressure is measured in pounds.

32. Nominal and Actual Braking Force.—Braking force may be either nominal or actual. The nominal braking force is a calculated force which cannot be obtained in practice, as the word *nominal* in this case means that no account is taken of the losses which occur in the transmission of brake-cylinder pressure to the brake shoes. These losses may be due to the effect of the release springs, the suspension of the brake hangers, the friction of the moving members of the brake rigging, and other causes. Tests have shown that the efficiency of a well-designed foundation brake rigging in transmitting brake-cylinder force to the brake shoes is about 85 per cent. In other words, 15 per cent. of the brake-cylinder force is lost in transmission.

The actual braking force takes into consideration all the losses that occur in the transmission of brake-cylinder pressure, and means the actual brake-shoe pressure obtained when any given brake application is made.

The nominal braking force is based on the braking force obtained from either a full-service or an emergency application. The actual braking force is that obtained from any service application or an emergency application. Unless otherwise stated, the term braking force refers to the nominal service-braking force.

The following will serve to show the difference between the nominal and the actual braking force: If, in a full-service application, there is a calculated pressure of 2,500 pounds against one brake shoe, and if there are eight shoes, the nominal braking force is $2{,}500 \times 8 = 20{,}000$ pounds. If a test showed that the pressure actually developed was 17,500 pounds, this would be the actual braking force.

33. Pressure on Which Braking Force is Based. The service-braking force for a freight car is based on a brake-cylinder pressure of 50 pounds and the service-braking force for the PM, LN, and UC equipments is based on a brake-cylinder pressure of 60 pounds. The service-braking force for

the PC equipment is based on a brake-cylinder pressure of 86 pounds. An emergency application with any of the equipments just mentioned will develop a higher brake-cylinder pressure. For example, an emergency application with freight equipment gives a brake-cylinder pressure of 60 pounds.

The emergency-braking force is then based on the brake-cylinder pressure obtained in emergency and is as follows: 60 pounds for freight equipment; 85 pounds for the PM equipment, which pressure gradually reduces to 60 pounds; 100 pounds for the LN equipment; and 100 pounds for the UC equipment. With the PC equipment, which uses two brake cylinders, the same pressure is obtained in each as in service, or 86 pounds. With later types of PC equipment, the emergency-brake cylinder is 2 inches less in diameter than the service-brake cylinder, and this reduces the brake-cylinder force obtained in emergency.

The higher brake-cylinder pressure in emergency with the LN and UC equipments is obtained by the use of an extra reservoir, which discharges to the auxiliary reservoir. The effect of this is to give a higher brake-cylinder pressure. The PC equipment uses an extra reservoir, as well as another brake cylinder, to obtain a high pressure.

BRAKING RATIO

34. Definition.—The meaning of the term *braking ratio* will be explained by the following example: If it is assumed that a car weighs 100,000 pounds when empty and that the braking force is 60,000, a comparison between the braking force and the weight of the car is made by dividing the braking force by the weight. Thus, $\frac{60000}{100000} = \frac{6}{10}$, or 60 per cent. Therefore, a comparison between the braking force and the weight shows that the former is 60 per cent. of the latter.

The term braking ratio is applied to the comparison between the braking force and the weight of a car when empty, expressed in per cent. The reason for the use of the term braking ratio is that it enables a person to calculate the braking force when the empty weight of the car is known. In the

example given the braking ratio is 60 per cent. or .60, because a comparison between the braking force and the weight shows that the braking force is 60 per cent. of the weight. A braking ratio of 60 per cent. means that a braking force of 60 pounds is used for each 100 pounds of car weight. The braking force of freight cars is always less than the empty weight of the car, but with modern passenger cars, the braking force always exceeds the empty weight of the car.

The braking ratio is based on the weight of the car when it is empty, because if it were based on the weight of the car when it was loaded, the braking force would be so great when the car was unloaded that there would be danger of sliding the wheels.

The term formerly used for braking ratio was *per cent. braking power*, but, as already explained, the term power in this instance is incorrect. The term per cent. of braking force is also sometimes used with a meaning the same as braking ratio. However, as it is desired to compare the braking force with the light weight of the car, and as the word *ratio* means a comparison, the term braking ratio is the correct one to use.

35. Rule for Finding Braking Ratio.—The braking ratio can be found from the following rule:

Rule.—*To find the braking ratio, multiply the braking force by 100 and divide by the weight of the car when empty.*

EXAMPLE.—A car when empty weighs 50,000 pounds, and the braking force is 30,000 pounds. What is the braking ratio?

SOLUTION.—From the rule, the braking ratio equals

$$\frac{30,000 \times 100}{50,000} = 60 \text{ per cent. Ans.}$$

36. Finding Braking Force from Braking Ratio. As the braking ratio indicates the per cent. of the empty weight of a car which is used for the braking force, the braking force can be found from the following rule:

Rule.—*To find the braking force, multiply the empty weight of the car by the braking ratio.*

EXAMPLE.—A car weighs 120,000 pounds, and the braking ratio is 90 per cent. What is the braking force?

SOLUTION.—From the rule, the braking force equals 120,000×.90 (90%) or 108,000 lb. Ans.

37. Braking Ratio Affected by Loading.—As the braking force remains the same whether the car is empty or loaded, the braking ratio will be affected by loading the car. If a box car weighs 50,000 pounds when empty, and it is capable of carrying a load of 100,000 pounds, if the braking force is 30,000 pounds, the braking ratio when the car is empty is $\frac{30{,}000\times 100}{50{,}000}$, or 60 per cent. The total weight of the car when loaded is 150,000 pounds, but the braking force remains the same, or 30,000 pounds. The braking ratio is now equal to $\frac{30{,}000\times 100}{150{,}000}=20$ per cent. Therefore, the braking ratio decreases when the car is loaded.

The foregoing applies to the single-capacity brake. With the empty and loaded brake, the braking ratio is 60 per cent. when the car is empty and 40 per cent. when the car is loaded. The difference between the weight of a passenger car when it is loaded and empty, is small, and the braking ratio is not affected to any great extent by the load.

38. Nominal and Actual Braking Ratio.—As the braking force is either nominal or actual, the braking ratio must also be either nominal or actual. The nominal braking ratio is the relation between the nominal braking force calculated on a full-service or an emergency application, and the weight of the car when it is empty. The actual braking ratio refers to the relation between the braking force that results from any brake application and the weight of the car, whether it is loaded or empty.

Unless otherwise stated, the braking ratio means the nominal braking ratio with the braking force based on a full-service application.

39. Service-Braking Ratios.—In Table I is given the service-braking ratios that are used with the various car-brake equipments. It is often required to know the brake-cylinder pressure on which the braking force is based. Therefore, the brake-cylinder pressures from Art. **33** have been repeated. This table shows that the braking ratio for freight cars is lower than for passenger cars. The reason is that the speed of freight cars is slower, and the slower the speed, other things being equal, the more effective the brake shoes are in retarding the speed of the wheels.

TABLE I

SERVICE-BRAKING RATIOS FOR FREIGHT AND PASSENGER CARS

Equipment	Braking Ratio Per Cent.	Brake-Cylinder Pressure On Which Braking Force Is Based Pounds
Freight..........	60	50
PM.............	90	60
LN..............	90	60
PC..............	90	86
UC..............	90	60

The higher braking ratio on cars in passenger service is obtained by the use of larger brake cylinders and by using a higher brake-pipe pressure. The pressure of equalization is higher as the brake pipe and the auxiliary-reservoir pressures are increased.

40. Table II gives the braking ratio for locomotives when the ET and A-1 brake equipments are used, and also the brake-cylinder pressure on which the braking force is based.

TABLE II

SERVICE-BRAKING RATIOS FOR LOCOMOTIVES AND TENDERS

Equipment	Braking Ratio Per Cent.	Brake-Cylinder Pressure on Which Braking Force Is Based Pounds
ET (Passenger)	60	50
ET (Freight)	50	50
ET (Tender)(Passenger)	80	50
ET (Tender)(Freight)	70	50
ET (Truck and trailer)	45	50
A-1 (Passenger)	75	50
A-1 (Freight)	65	50
A-1 (Tender)(Passenger)	85	50
A-1 (Tender)(Freight)	80	50
A-1 (Truck and trailer)	60	50

41. Emergency-Braking Ratios.—In Table III are given the emergency-braking ratios that are used with the different car equipments, as well as the brake-cylinder pressure on which the braking force is based.

TABLE III

EMERGENCY-BRAKING RATIOS FOR FREIGHT AND PASSENGER CARS

Equipment	Braking Ratio Per Cent.	Brake-Cylinder Pressure On Which Braking Force Is Based. Pounds
Freight	72	60
PM	90	85
LN	150	100
PC	150	86
UC	150	100

With freight equipment, the service-braking ratio is 60 per cent. with 50 pounds pressure. Therefore, the emergency-braking ratio is 72 per cent. with 60 pounds pressure. However, 70 per cent. is usually taken as the emergency-braking ratio. The braking ratio of the PM equipment is 125 per cent. with a brake-cylinder pressure of 85 pounds. However, as this pressure blows down to 60 pounds, the braking ratio is given as 90 per cent. The calculated emergency-braking ratio with the LN equipment and 104 pounds brake-cylinder pressure is 156 per cent. However, the single-brake-shoe rigging is generally used with this equipment, and as this causes a reduction in brake-cylinder pressure on account of false piston travel, the braking ratio will be less than the amount given. Therefore, the braking ratio is given as 150 per cent. with a brake-cylinder pressure of 100 pounds. With the PC equipment, it is assumed that the emergency brake cylinder is smaller than the service cylinder, and the emergency-braking ratio is accordingly reduced from 180 to 150 per cent.

42. Service- and Emergency-Braking Ratios.—It will be noted that the emergency-braking ratios are much higher than the service-braking ratios. The reason is that the smoothness of the stop is not taken into account in emergency, the only consideration at this time being the development of sufficient braking force to stop as quickly as possible so long as this force does not exceed the adhesion between the wheels and the rail.

The reason the higher emergency ratios given do not cause the wheels to slide is that the frictional force of the brake shoes at high speed is less than the adhesion between the wheel and the rail. This is explained in Art. **22.** Tests have shown that the sliding of wheels is influenced more by the condition of the rails than by a high brake-shoe pressure. At low speed the coefficient of friction is high and an emergency application with a high braking ratio will then probably cause the wheels to slide.

43. Difference Between Rail Friction and Brake-Shoe Friction.—The following example shows the differ-

ence between the rail friction and the brake-shoe friction when a high-emergency braking ratio is used:

EXAMPLE.—A car weighs 120,000 pounds and has a braking ratio of 150 per cent. What is the difference between the frictional resistance at the rail and the wheel when the average coefficient of friction is 15 per cent., and the coefficient of adhesion 25 per cent.?

SOLUTION.—The braking force is equal to 120,000×150 per cent., or 1.5=180,000 lb. The friction developed by the brake shoes is equal to the braking force multiplied by the coefficient of friction, or 180,000×.15 =27,000 lb.

The friction between the wheel and the rail is equal to the weight of the car multiplied by the coefficient of adhesion, the friction is, therefore, equal to 120,000×.25=30,000 lb. The brake-shoe friction is equal to 27,000 lb. The difference between 30,000 lb., the rail friction, and 27,000 lb., the wheel friction, is 3,000 lb., which is the excess to keep the wheel turning. Ans.

PISTON TRAVEL

EFFECT AND ADJUSTMENT

44. Definition.—By the term *piston travel* is meant the distance the brake-cylinder piston travels from release position when a full-service application is made.

45. Necessity of Uniform Piston Travel.—In order to obtain the same braking force or brake-shoe pressure, it is essential that the piston travel shall be the same on all cars.

The brake-shoe pressure is the result of the brake-cylinder pressure and the pressure developed in the brake-cylinder depends on the brake-cylinder volume into which the air from the auxiliary reservoir expands when an application of the brake is made. The brake-cylinder volume depends on the piston travel and it increases with the travel. Therefore, when the piston travel is long, the brake-cylinder pressure will be less than when the travel is short, because the amount of air that enters the brake cylinder is dependent on the brake-pipe reduction. Therefore, the same amount of air passes to the brake

cylinder whether the piston travel is long or short. The braking force will be greater on a car with a short travel, and this car, when the brakes are applied, will stop sooner than a car on which the travel is longer. Unequal piston travel is then undesirable, as it causes shocks when trains are stopped on account of the unequal braking forces that are developed on the different cars.

Unequal braking forces on the cars are also liable to cause the wheels to slide when the brakes are applied. When the brakes are applied the cars with the high-braking force will stop quicker than the cars with the low braking force. The cars with the low braking force then transmit a push or a pull to the cars with the high braking force. For an instant the cars with the high braking force will be moving faster than the speed at which their wheels are revolving, and at such a time the wheels are liable to slide.

46. Standing and Running Piston Travel.—The standing piston travel is the piston travel when the car is not in motion and the running piston travel is the travel when the car is moving. The running piston travel is always more than the standing piston travel. The reason is that the lost motion that exists in the truck-center bearings is more readily taken up when the car is moving than when it is standing. The running travel is generally from 1 to 3 inches greater than the standing travel.

The standard piston travel for all forms of car brakes is 8 inches, and this means the running travel, because if the standing travel were meant, the running travel would be more than 8 inches. The practice is then to adjust the standing travel so that the proper running travel will be obtained when the car is moving. This requires that the standing travel be about $6\frac{1}{2}$ inches, and a standing travel of this amount will give a running travel of about 8 inches.

47. False Piston Travel.—By the term *false piston travel* is generally meant the difference between the running and the standing travel. But, strictly speaking, it means the difference in piston travel caused by the springing of the brake

beams, levers, and other parts of the foundation brake rigging and the pulling down of the truck frames, and, therefore, of the brake shoes. False piston travel is, then, due to poor foundation brake-rigging design, and it increases with the brake-cylinder pressure.

THE AMERICAN AUTOMATIC SLACK ADJUSTER

48. Purpose.—The purpose of the American automatic slack adjuster is to maintain automatically a uniform running piston travel of 8 inches. A piston travel in excess of this amount due to brake-shoe wear or other causes, is shortened 1/30 inch each time the brake releases. Slack adjusters are applied only to passenger cars.

49. Construction.—The construction of the automatic slack adjuster will be explained by referring to Fig. 17. View (*a*) shows that the slack adjuster is attached to a projection on the pressure head of the brake cylinder by two bolts and nuts *a*. The crosshead *b*, which operates between the two guides shown, is connected to the end of the dead-cylinder lever by a bolt. The bolt may be removed and replaced by taking off the cap nut b_1. The threaded adjuster rod b_2 is attached to the crosshead, and this rod extends back through an opening in the part b_3 which is cast with the guides, and also through a hole in the casing *c*. This casing is bolted to the part b_3, and it is shown partly broken so that the interior may be seen. A ratchet nut b_4 is screwed on to the outer end of the adjuster rod and the inner face of this nut bears against the outer face of the part b_3. The ratchet nut turns freely where it passes through the casing *c*. The ratchet nut differs in shape from the usual form of nut inasmuch as it has a hollow extension b_5 which protects the thread on the adjuster rod, and it also has a series of teeth cut around the outside of its inner end as shown. It is for this latter reason, that the nut is called a *ratchet nut*. The part of the nut with its teeth is enclosed by the casing *c*.

The adjuster cylinder *e* which is shown in section in views (*b*) and (*c*), is bolted to the casing *c*. The cylinder, view (*b*), contains a piston e_1 with a packing leather, spring, and fol-

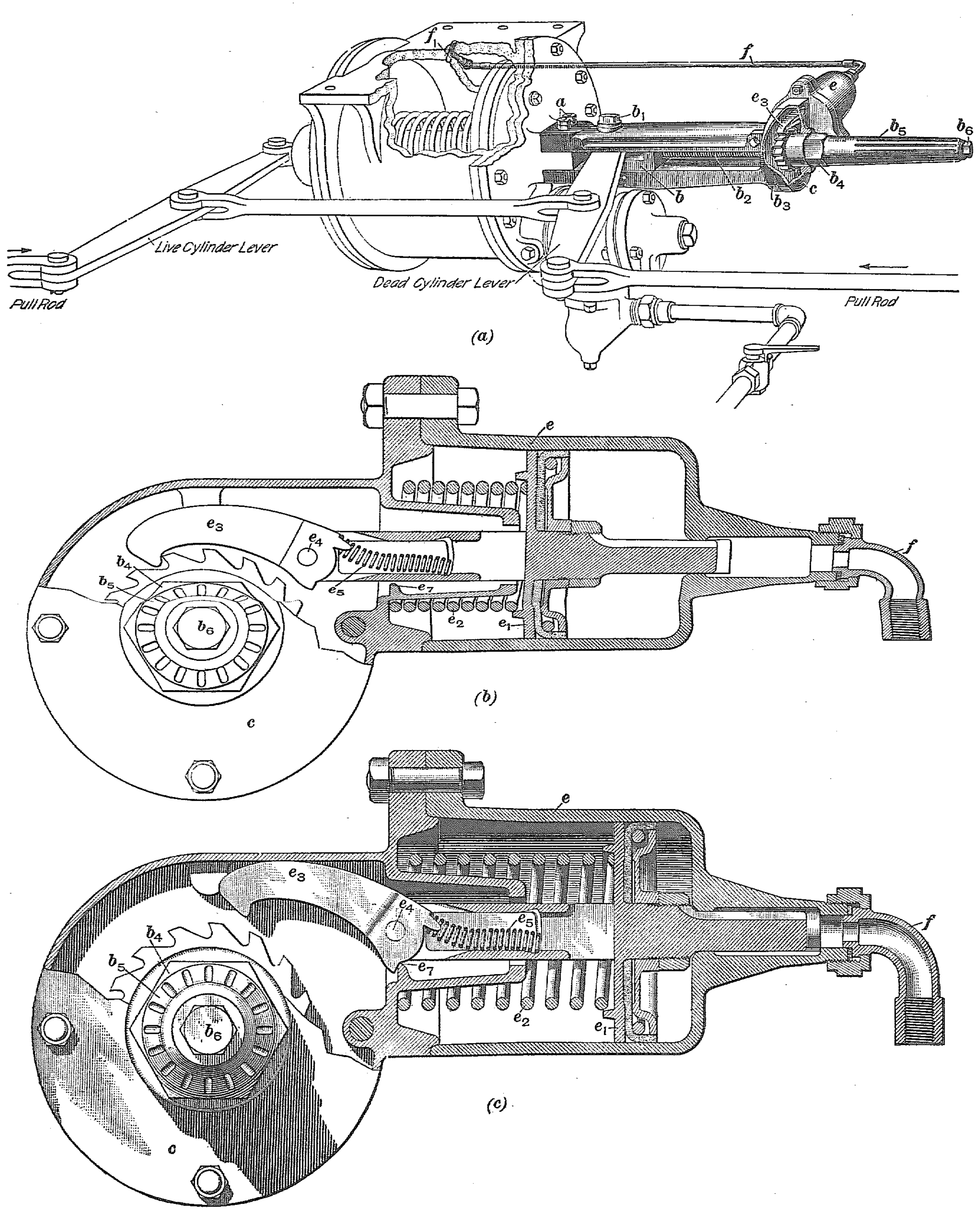
Live Cylinder Lever
Dead Cylinder Lever
Pull Rod
Pull Rod
(a)
(b)
(c)

FIG. 17

lower, a spring e_2, a pawl e_3 which is connected by a pin e_4 to the piston stem, and a pawl spring e_5. The pawl e_3 extends into the casing c above the ratchet nut. A pipe f, view (a), connects the adjuster cylinder to a hole f_1 which is tapped into the brake cylinder. The air in the brake cylinder can enter port f_1 when the brake-cylinder piston moves out about 8 inches.

50. Operation.—The following preliminary explanation will serve to make the operation of the slack adjuster more readily understood. If we assume that the ratchet nut is turned to the right by applying a wrench to the flattened surfaces of the nut, the ratchet nut cannot be screwed forwards on the rod because the end of the nut bears against the inner face of the part b_3 of the adjuster. Therefore, when the nut is turned the rod will be drawn outwards through the hollow ratchet nut. The crosshead and the end of the dead cylinder lever will move in the same direction as the rod. The pull rods will be moved in the direction of the arrows, and the brake shoes will thereby be brought nearer the wheels.

The manner in which the ratchet nut is turned by the air pressure is as follows: When the piston in the brake cylinder moves far enough to allow the packing leather to pass the port f_1, the air from the brake cylinder passes through pipe f to the adjuster cylinder e. The adjuster piston e is forced forwards, view (b), and the pawl e_3 which is forced downwards by the spring e_5, engages one of the teeth of the ratchet nut. The pawl moves over two teeth of the ratchet nut each time the adjuster operates. When the brake is released and the brake-cylinder piston moves back past port f_1 the air in the adjuster cylinder passes to the atmosphere through the non-pressure head of the brake cylinder. The spring e_2 in the adjuster cylinder forces the piston e_1 back and the pawl turns the ratchet nut about one-eighth of a turn. This shortens the piston travel one-thirtieth of an inch. As the take-up movement is almost completed the pawl e_3 engages the projection e_7 view (c) and the pawl is thrown out of contact with the ratchet nut. As one application of the brakes shortens the piston travel one-thirtieth of an inch, it will require thirty brake applications

to shorten the travel 1 inch. This requires that the ratchet nut be turned about four times.

51. Letting Out the Slack.—The adjuster is only automatic when the slack is being taken up. The slack must always be let out by hand. This is done by turning the ratchet nut to the left with a wrench. When it is desired to let out the slack the ratchet nut should be always moved slightly to the right before it is turned to the left. This precaution is necessary to free the pawl in the event of its being stuck in a tooth of the ratchet. The teeth or the pawl would be damaged were an attempt made to turn the ratchet nut to the left when the pawl is engaged with the nut.

52. Purpose of Stop Screw.—The purpose of the stop screw b_6 is to enable one to let out the slack after it has been all taken up. When the slack is all taken up, the end of the threaded rod is up against the stop screw, and the pawl remains engaged with the ratchet nut, because the nut cannot now move the rod any farther. The pawl can be freed by removing the stop screw, as this permits a sufficient take-up movement to free the pawl. The slack can then be let out.

53. Replacing Brake Shoes.—When new brake shoes are to be applied the slack should be let out, as already explained. If the slack is all taken up one should proceed as explained in Art. **52.** Sufficient slack should be let out to permit of at least $8\frac{1}{4}$ inches piston travel after the new shoes are applied. After the shoes have been applied, it is necessary to take up the slack so that the standing piston travel is $6\frac{1}{2}$ inches, because with this standing travel the running travel will be about 8 inches. To do this, the brakes should be set in full, and the piston travel measured. If the piston travel is found to be 9 inches, the brake should be released and the ratchet nut turned to the right until the slack-adjuster crosshead moves $2\frac{1}{2}$ inches toward the adjuster cylinder. The standing travel will then be $6\frac{1}{2}$ inches.

54. Adjuster Takes Up too Much Slack.—The primary purpose of the slack adjuster is to prevent the running

piston travel from exceeding a certain amount, due to the wear of the brake shoes or to the lost motion in the brake-rigging connections. However, any other condition which may bring about a long piston travel, will also cause the adjuster to operate. As will be explained later, a single-shoe type of foundation brake rigging causes an excessive piston travel when the brakes are applied while the car is in motion, especially if the braking ratio is high. The car may start out with the correct standing travel, but as the running travel is so much more, the adjuster will operate at each brake application. As a result, when the car reaches a terminal the standing piston travel may be so short that the brake shoes will not clear the wheels. Therefore, when the slack adjuster takes up too much slack, it does not imply any disorder in the adjuster. Such a condition points out the necessity of a proper design of brake rigging.

55. To Take Up Slack on Freight Equipment. The slack in freight-brake equipments is taken up by hand because these equipments are not supplied with slack adjusters. The dead truck levers are secured in the guides by pins. When necessary the slack is taken up by removing the pins and moving the levers to holes that are nearer the brake cylinders. The slack is taken out by moving the ends of the levers in the opposite direction in the guides. The parts of the brake beams through which the truck levers pass sometimes have holes for the purpose of adjusting the slack. The slack should be taken up equally at each truck.

LEVERAGE RATIO

56. The *leverage ratio* of a single lever is found by dividing the force that is delivered by the force that is applied. For example, if the applied force is 10 pounds and the delivered force is 50 pounds, the leverage ratio is said to be 5 to 1. It then follows from the principles of levers that the length of the lever between the applied force and the fulcrum is five times as long as the length of the lever between the delivered

force and the fulcrum. The point at which the force is applied will then move five times as far as the end at which the force is delivered.

The foregoing also applies to a system of levers such as the foundation brake rigging. If it is assumed that a brake-cylinder force of 4,000 pounds delivers when it is transmitted through the brake levers, a brake-shoe pressure of 32,000 pounds, the result obtained when 32,000 is divided by 4,000 is 8, and the leverage ratio is 8 to 1. A leverage ratio of 8 to 1 as applied to the brake then means that the brake-cylinder force is multiplied eight times by the levers of the rigging before it is transmitted to the brake shoes. In this case the brake-cylinder piston will move eight times as far as the brake shoes.

57. Reason for not Using a High-Leverage Ratio. It is not always apparent why the brake-cylinder force cannot be increased to any extent desired by the use of levers. The reason will be understood when it is remembered that an increase of force by the use of a lever always implies that the end of the lever at which the force is applied must move farther than the end at which the force is delivered. In other words, a gain in force is accomplished only at the expense of distance. Therefore, if the brake-cylinder force is greatly increased by the brake rigging the brake cylinder would have to be made longer so as to permit the end of the lever where the force is applied, to move the required amount. A larger brake cylinder would require a larger auxiliary reservoir and also a larger main reservoir. In order to keep the size of the brake equipment within reasonable limits, it is necessary to limit the leverage ratio which is used. The usual leverage ratio is 8 to 1 or 9 to 1. Two brake cylinders are used if a greater brake-shoe pressure is desired than can be developed from these ratios.

58. In Figs. 18 and 19 are shown the effect of using a high-leverage ratio. These figures assume that the brake rigging consists of a single lever and a brake shoe. The piston travel in Fig. 18 is 8 inches and the leverage ratio is 8 to 1 because the arm *ab* is eight times as long as the arm *bc*. From

the principles of levers the brake-cylinder piston must move the end *a* 8 inches in order to move the point *c* or the brake shoe 1 inch. If the piston travel is 8 inches the brake shoe will stand 1 inch from the wheel, or the brake-shoe clearance will be 1 inch when the brake is released.

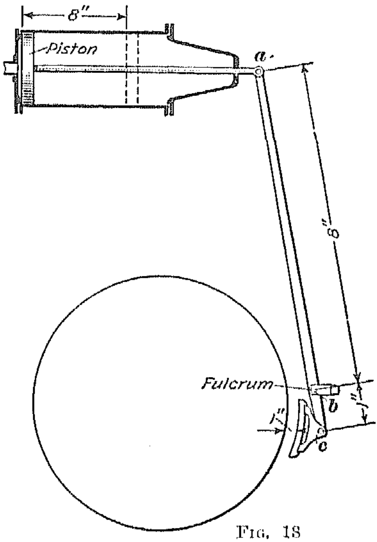

FIG. 18

In Fig. 19 the leverage ratio is 16 to 1 and the piston travel is 8 inches. The end *a* of the lever must now move 16 inches in order to move the brake shoe 1 inch. As the movement of the end *a* is restricted to 8 inches, the brake shoe will move ½ inch when the brake is applied. The brake-shoe clearance is then reduced to ½ inch when the leverage ratio is doubled. The brake-shoe clearance may be increased by simply hanging the brake shoe farther from the wheel. If the brake shoe is hung ½ inch farther from the wheel, making the total shoe clearance 1 inch, the piston would have to move 16 inches before the brake shoe would strike the wheel, and the length of the brake cylinder is limited to 12 inches.

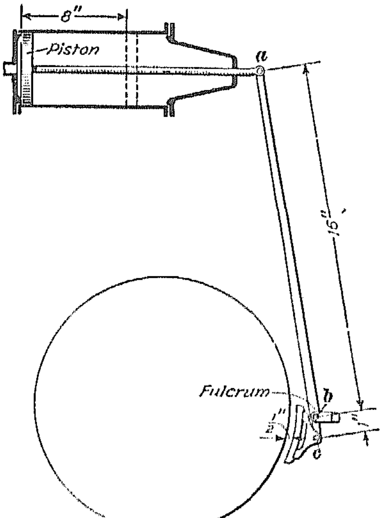

FIG. 19

59. When the leverage ratio is high, the piston travel will increase more on account of brake-shoe wear or other con-

ditions which may cause an increase in travel, than when the leverage ratio is low. Thus, when the leverage ratio is 16 to 1 the piston travel will increase twice as much as when the ratio is 8 to 1. If the ratio is 16 to 1 and the brake-shoe wear is $\frac{1}{8}$ inch, the brake-cylinder piston will move sixteen times $\frac{1}{8}$ inch or 2 inches farther than before, whereas when the ratio is 8 to 1 the piston will move eight times $\frac{1}{8}$ inch, or 1 inch farther.

Therefore, a high-leverage ratio cannot be used when the piston travel is limited to a fixed amount, because the result would be a reduced shoe clearance, and excessive piston travel from brake-shoe wear, springing of the brake rigging, journals moving in their brasses or slack in the center bearings.

60. Brake-Shoe Clearance.—The brake-shoe clearance can easily be found when the leverage ratio is known. If the leverage ratio is 8 to 1, the brake-cylinder piston must move 8 inches in order to move the brake-shoe 1 inch. Therefore, when the piston travel is 6 inches, the brake shoes move $6/8$ or $\frac{3}{4}$ inch, which will be the shoe clearance. The shoe clearance can then be found from the following rule:

Rule.—*To find the shoe clearance, divide the piston travel by the leverage ratio.*

61. Finding Braking Force From Leverage Ratio. The braking force can be found when the leverage ratio and the brake-cylinder force is known. For example, a pressure of 50 pounds in a brake cylinder that is 10 inches in diameter, will give a force of 3,950 pounds on the brake-cylinder piston. If the leverage ratio multiplies this force eight times, the nominal or calculated service-braking force would be equal to $3{,}950 \times 8 = 31{,}600$ pounds.

FOUNDATION BRAKE RIGGING

(PART 2)

Serial 2071B Edition 2

ACTION AND CALCULATION OF BRAKE-RIGGING FORCES—(Continued)

BRAKE RIGGING

1. Purpose and Requirements.—The purpose of the foundation brake rigging is to increase and transmit to the wheels equally the force exerted by the compressed air in the brake cylinder. Therefore the foundation brake rigging forms the connecting link between the brake cylinder and the wheels, and converts the air pressure in the cylinder into mechanical force on the wheels. It consists of a system of levers, rods, pins, hangers, brake beams, and brake shoes.

The first and essential requirement of the foundation brake rigging is that it be designed with due regard to strength, rigidity, and arrangement so as to insure a piston travel as nearly constant as possible under all variations in brake-cylinder pressure. The brake rigging should be designed also so that the pressure applied to the wheels will not force the journals from under their bearings and cause journal troubles.

TYPES OF BRAKE RIGGING

2. Names of Types.—Foundation brake rigging may be classed as of two general types, according to the number of brake shoes that are used per wheel. The *single-shoe-per-wheel type* uses one brake shoe for each wheel, and is universally used on freight cars and the older passenger cars. The *two-shoe-*

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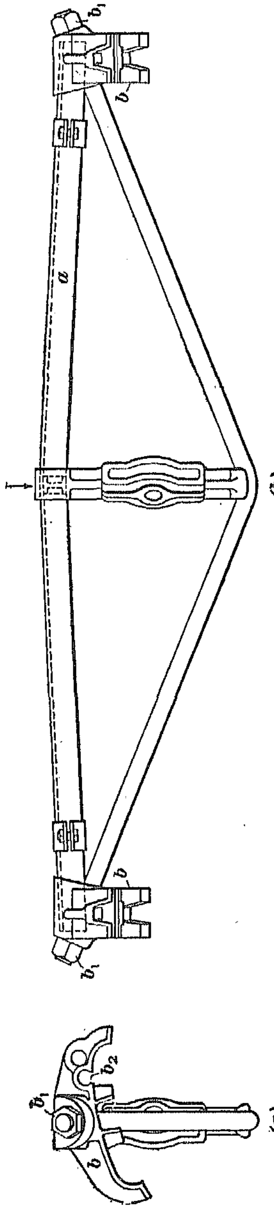

FIG. 2

per-wheel type uses two brake shoes for each wheel and is in universal use on modern passenger cars. This type of brake rigging is commonly known as the *clasp brake.*

SINGLE-SHOE-PER-WHEEL TYPE OF BRAKE RIGGING

3. Description.—In Fig. 1 is shown a single-shoe-per-wheel type of brake rigging for a freight car, with the principal parts named. The live cylinder lever is pinned to the brake-cylinder piston. One end of this lever is pinned to the hand-brake rod and the other end is pinned to the top rod. The cylinder-lever rod connects the live cylinder lever to the floating lever. One end of the floating lever is connected by a pin to the floating-lever bracket and the other end of the lever is pinned to the top rod, which is connected to the upper end of the live truck lever and the lower end of the lever is pinned to the bottom rod. The brake beam is connected to the live truck lever between its ends. The bottom rod is connected by means of a pin to the lower end of the dead truck lever and the upper end of this lever is pinned to the dead lever guide. The brake beam is pinned to the dead truck lever between the ends. The arrangement of the brake rigging is the same on the other truck. The brake beams are suspended by brake hangers to a

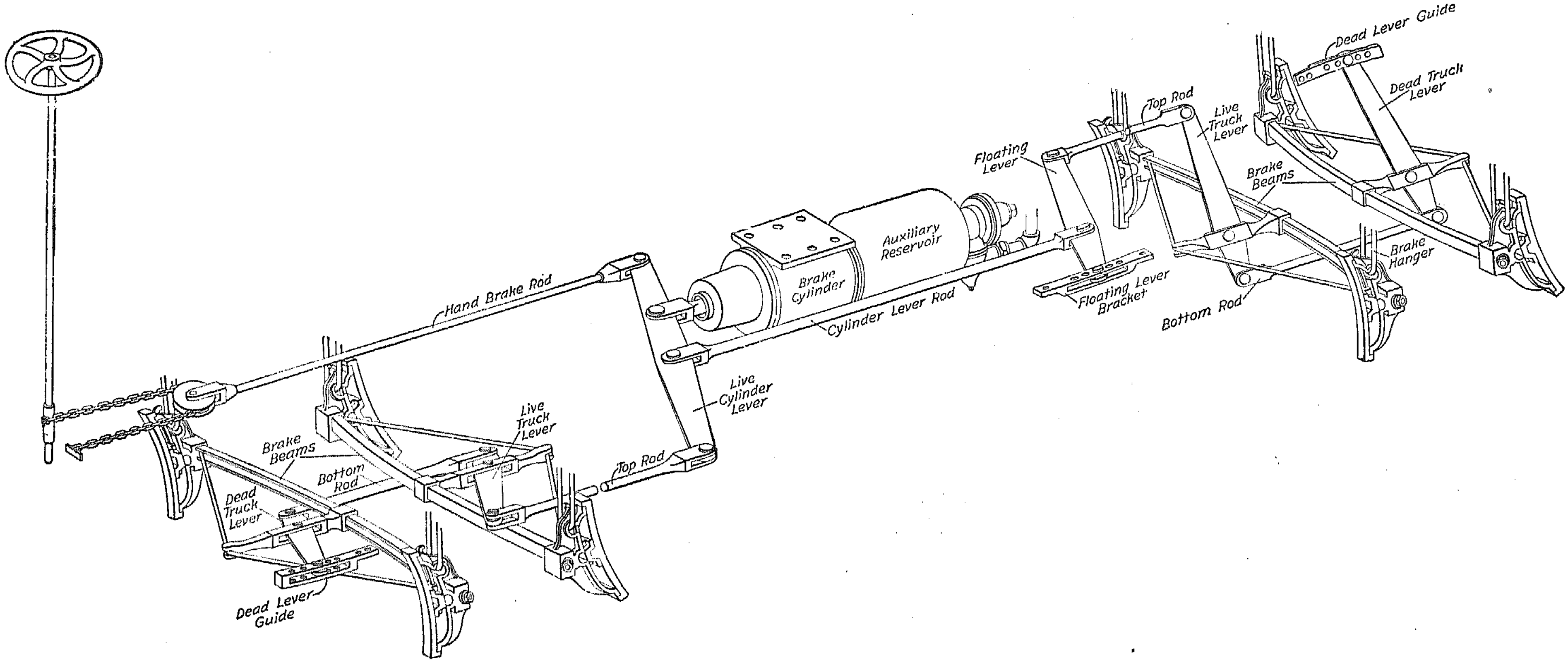

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Fig. I

part of the car truck. The floating-lever bracket is connected to the car sill and the dead lever guides are connected to the truck bolster.

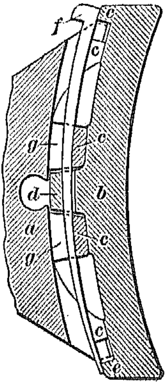


FIG. 3

4. Details of the Parts.—Two views of a brake beam are shown in Fig. 2. The brake heads b are secured to the brake beam by nuts b_1. The brake hangers rest in a semicircular recess b_2, view (*a*) in the brake head. The hangers are held in place by the brake shoes when they are attached to the brake head. The brake beam shown in Fig. 2 is only one of the many types in use.

A vertical section of a brake shoe and a part of the brake head is shown in Fig. 3, and illustrates how the brake shoe is secured to the head. The brake head *a* has four contact surfaces *c*, and the brake shoe is held against these surfaces by the brake-shoe key *f*. This key passes through the eyes *g* in the brake head and the eye *d* in the shoe. The brake-shoe key also makes a contact with the shoulders *e* of the brake shoe. The brake hanger rests in the semicircular recess shown in the brake head.

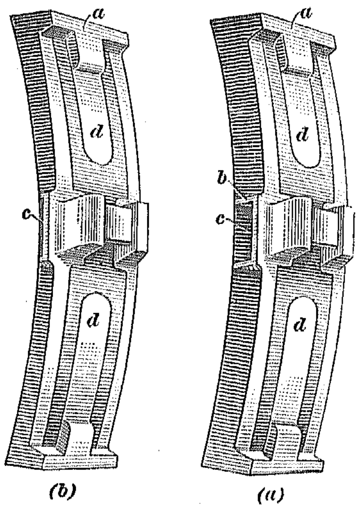


FIG. 4

In Fig. 4 (*a*) and (*b*) are shown two views of a brake shoe. The brake shoe has a mark at *b* similar to a ⌈‾‾‾‾⌉ section cast on the side of the shoe. When the shoe is worn down where only the portion *c* of the mark remains, as in view (*b*), the brake shoe must be discarded. The portions marked *d* are inserts of mild steel which serve to retain the pieces in

position in case the shoe is fractured. The brake shoes used on passenger cars are usually flanged so that they engage not only the tread but also the flange of the wheel.

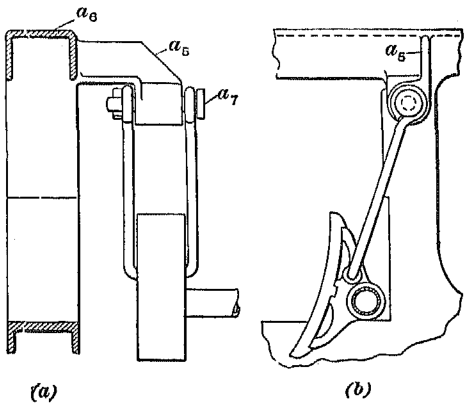


FIG. 5

Fig. 5 (*a*) and (*b*), shows how the brake beams are suspended to the truck when the side frame of the truck is in one piece. A brake-hanger carrier a_5 is cast on the side frame a_6 of the truck and the brake hanger is connected to it by the brake hanger pin a_7. In Fig. 6 (*a*) and (*b*) is shown the method of suspending the brake beams with an arch-bar truck. The brake-hanger carrier is cast with the column casting or bolster guide, as shown, and the brake hanger is pinned to it.

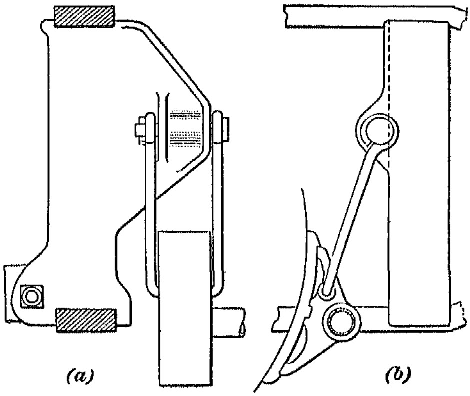


FIG. 6

With either of these arrangements the loading or unloading of the car does not affect the position of the hangers and the brake beams, as the hanger carriers are not affected by the movement of the truck bolster.

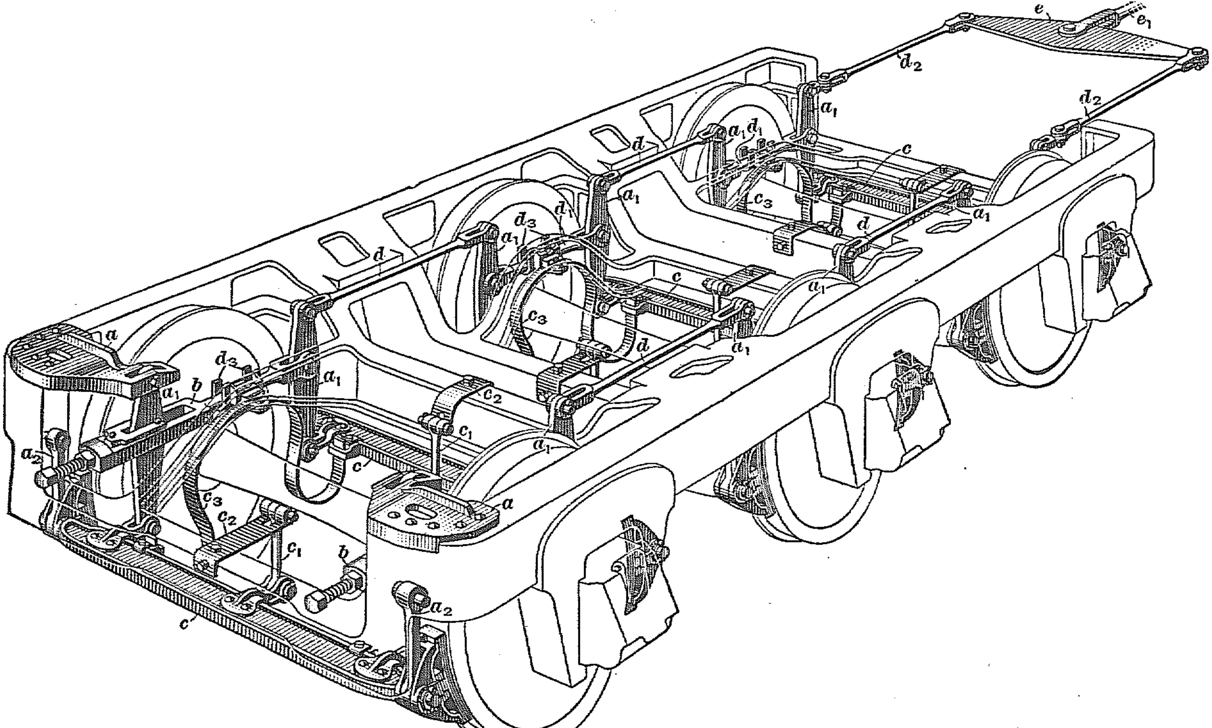

e
e_1
d_2
d_2
a_1
a_1 d_1
d
c
a_1
c_3
a_1
d
d_1
d_3
a_1
d
c
a_1
c_3
d
a_1
a
d_3
a_1
b
a_1
c_2
c_1
a_1
a_2
c
a
c_3
c_2
c_1
b
a_2
c

FIG. 7

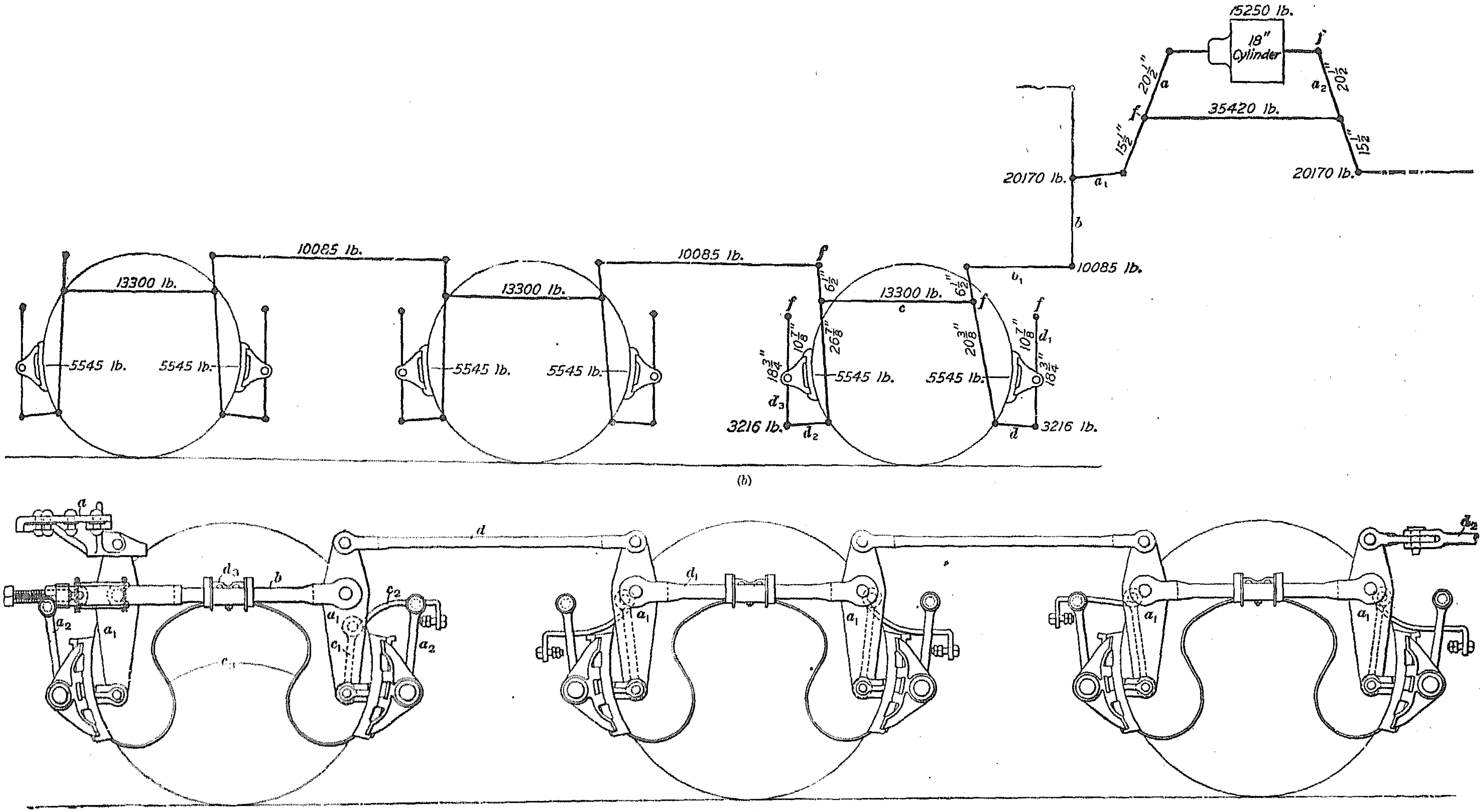
THE TWO-SHOE-PER-WHEEL TYPE OF BRAKE RIGGING (CLASP BRAKE)

5. Description.—In Fig. 7 is shown the clasp-brake type of brake rigging as applied to a six-wheel truck. Fig. 8, view (*a*), shows the brake rigging as viewed from the side. View (*b*) is an outline of the gear and shows the stresses on the different points. This view will be referred to when the calculation of brake-rigging forces are taken up later on.

The names of the parts indicated in Figs. 7 and 8 are as follows: a, dead lever fulcrum; a_1, truck lever; a_2, brake hanger; b, slack-adjuster pull rod; c, brake beam; c_1, balance hanger; c_2, balance-hanger bracket; c_3, release spring; d, top truck rod; d_1, bottom truck rod; d_2, truck connecting-rod; e, horizontal equalizer; e_1, horizontal equalizer connecting-rod.

The lower ends of the truck levers, a_1, Fig. 7, are pinned to brackets riveted to the brake beams. The balance-hanger brackets c_2 are secured to the truck frame, and the balance hangers c_1 are pinned to the brackets and to brackets riveted to the brake beams. These hangers prevent the brake beams from tipping and thereby keep the brake beams in their proper position relative to the wheels. The release springs c_3 keep the brake shoes from dragging on the wheels when the brakes are released. These springs are clamped to the pull-rod guides d_3. The horizontal equalizer connecting-rod e_1 connects to the lower end of the live cylinder lever.

6. Slack Adjuster.—The slack adjuster, which is shown on the end of the slack-adjuster pull rod, Fig. 7, is an arrangement whereby the slack in the brake rigging may be taken up or let out as desired. The construction of the slack adjuster is shown in Fig. 9, views (*a*), (*b*), and (*c*). View (*a*) shows the slack adjuster as viewed from the top; view (*b*) shows the adjuster as viewed from the side; and view (*c*) shows the crosshead. The slack adjuster, view (*a*), consists of a copper plated threaded bolt a, a jamb nut a_1, a crosshead b which works in a slot in the slack-adjuster pull rod, two cotter keys b_1, one of which is shown in view (*b*), a cotter key b_2, and a cotter key b_3. The truck lever extends through the slot in the crosshead,


15250 lb.
18" Cylinder
35420 lb.
20170 lb.
20170 lb.
10085 lb.
10085 lb.
10085 lb.
13300 lb.
13300 lb.
13300 lb.
5545 lb.
3216 lb.
3216 lb.

(b)

(a)

Fig. 8

view (*a*), and this lever is secured to the crosshead by the pin *c*. As shown in views (*b*) and (*c*), a slot is provided in the front end of the crosshead, and the pin *c* fits in this slot and is held in place by two cotter keys b_1 which extend down through the crosshead in front of the pin. The key b_2, view (*b*), extends through the crosshead and the threaded bolt *a*, and in combination with jamb nut a_1, serves to keep the bolt from turning. The key b_3 extends through the bolt *a* and prevents the bolt from being lost in the event of the failure of the jamb nut and pin b_2. The crosshead is placed in the slot in the adjuster rod by making the rod in two parts. The crosshead is then placed in the slot and the two parts are welded together.

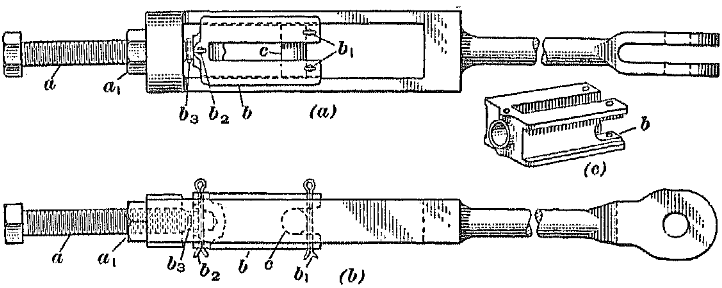


FIG. 9

The slack may be taken up by removing the key b_2, loosening the jamb nut a_1, and turning the threaded bolt to the right, and let out by turning the bolt to the left. The bolt is then turned to a position in which the key b_2 may be replaced, and the jamb nut is then tightened. If it becomes necessary to remove the slack-adjuster pull rod, as much slack as possible is let out and the two keys b_1 are removed. The crosshead is then moved to the rear until it is clear of the truck lever. The slack-adjuster pull rod may then be removed from the truck lever by disconnecting the upper end of the lever from the bracket.

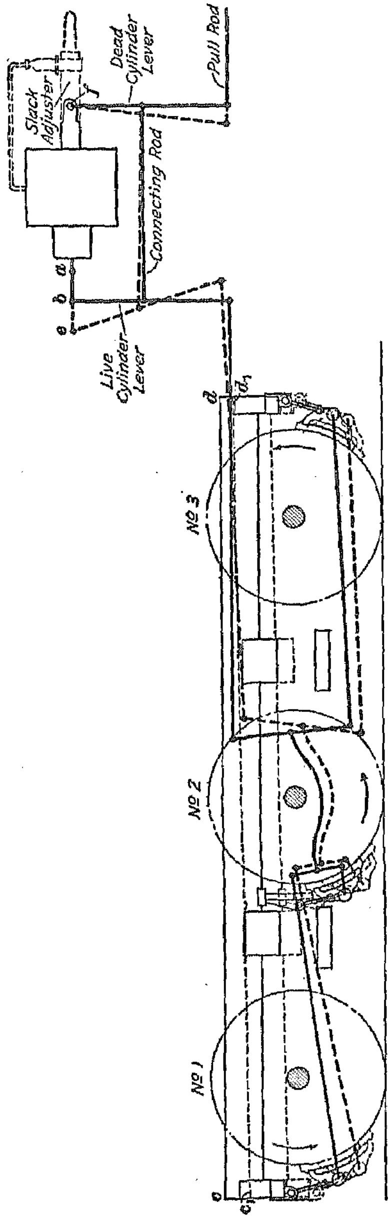


Fig. 10

TYPES OF BRAKE RIGGING COMPARED

7. Disadvantages of Single-Shoe-per-Wheel Type.—The principal objection to the single-shoe-per-wheel type of brake rigging is the tilting of the trucks and the consequent increase in piston travel and reduction in brake-shoe clearance that attends its use when the braking ratio is high. While this type of brake rigging is entirely unsuited to use on passenger cars with a high braking ratio, it can be employed satisfactorily in cases where the braking ratio is low, as on freight cars.

8. Action of Brake Shoes in Tilting Truck. The action of the brake shoes in causing the truck to tilt when the brakes are being applied, will be understood by referring to Fig. 10. This figure shows a six-wheel truck which is moving in the direction indicated by the arrow, and which has one brake shoe per wheel. It will be noted that the brake shoes and brake rigging which

are shown in their normal position by full lines, are hung considerably below the center of the wheels. This is necessary when but one brake shoe is used, because if the shoes were hung higher up the heavy unbalanced force against one side of the wheel would push the journals out of their bearings. However, there is a heavy lateral thrust on the journal boxes even with the shoes hung as shown.

The position of the brake-cylinder piston when it is in release position is shown by *a*, Fig. 10, and *b* shows the position of the piston when the brakes are applied. The effect on the brake shoe on the No. 3 wheel will be considered first. The pressure against this brake shoe when it is hung low on the wheel forces it down under the wheel with a wedging action that has a tendency to lift the wheel off the rail. The wedging action of the brake shoe is opposed by the friction between the brake shoe and the wheel which tends to move the brake shoe upwards. As the brake shoe is not pulled directly against the center of the wheel, the wedging action exceeds the friction of the brake shoe. The brake shoe and the brake rigging then move down to their dotted line positions and the truck moves from *d* to d_1.

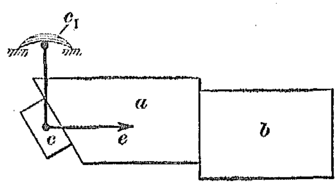


FIG. 11

The action of the brake shoe in pulling the truck down can be more readily understood by referring to Fig. 11, which shows a block *a* resting against a fixed part *b*, and a block *c* that is suspended by a hanger from a spring c_1. The block is pulled against *a* by the rod *e*. The block *c* may be compared to the brake shoe and the block *a* to the wheel. On account of the angle at which *a* is cut, the pull on the rod causes a wedging effect as well as friction between *a* and *c*. The block *c* will be pulled down, which shows that the wedging action of the block *c* is in excess of the friction between *a* and *c*.

The same action as was described at the No. 3 wheel, Fig. 10, also occurs at the brake shoes at No. 1 and No. 2 wheels, with the exception that the brake-shoe friction now assists the wedging action of the brake shoes. Therefore the brake shoes

on these wheels, as shown by their dotted outlines, will be pulled down further than on the No. 3 wheel. The left end of the truck will also move down farther, or from c to c_1. The line $c\ d$ shows the original position of the truck and the line $c_1\ d_1$ shows its position after the brake is applied. The downward movement of the truck puts a heavy overload on the equalizers, equalizer coil springs, and the axles.

9. Effect of Truck Tilting on Piston Travel.—The piston travel will increase when the truck tilts and the brake shoes move down under the wheels because the brake shoes are now moving closer to the brake cylinder. The brake-cylinder piston, instead of remaining at b, will move out to e. This movement of the piston gives a false piston travel equal to $b\ e$ and thereby lessens the pressure in the brake cylinder. If the piston travel $a\ e$ exceeds the travel permitted by the slack adjuster, the adjuster will operate and when the brakes are released will move the end f of the dead-cylinder lever outwards. The connecting-rod then pulls the live cylinder lever and point e closer to the brake cylinder and moves the brake shoes closer to the wheels.

When the brakes are applied again the same action will occur, that is, the brake shoes will pull down on the wheels and cause the piston travel to increase, and the slack adjuster will operate and bring the brake shoes closer to the wheels and thus reduce the piston travel. The final result will be that the piston travel instead of being maintained equal to $a\ b$ will become much less, or the piston, when the car is standing, may have to move only a few inches to bring the brake shoes up against the wheels. For example, if the piston travel were reduced to 3 inches and the leverage ratio is 9 to 1, the shoe clearance would be $\frac{3}{9}$, or $\frac{1}{3}$, inch, which would not be enough to keep the brake shoes clear of the wheels. When the piston travel is short it will be impossible to apply the brakes lightly without severe shocks on account of the high brake-cylinder pressure developed by the short travel.

The brake is referred to as not being flexible, when light brake applications cannot be made without causing shocks.

The foregoing shows that a single brake shoe cannot be hung low without a great loss of braking force. Neither can it be hung too high without danger of pushing the axle out from under the bearing. The reduction in shoe clearance caused by the slack adjuster suggests dispensing with the device altogether. However, were it not for the adjuster the brake-cylinder piston might move out far enough to strike the cylinder head. The operation of the slack adjuster is then as it should be and its action shows the necessity of changing the design of the brake rigging.

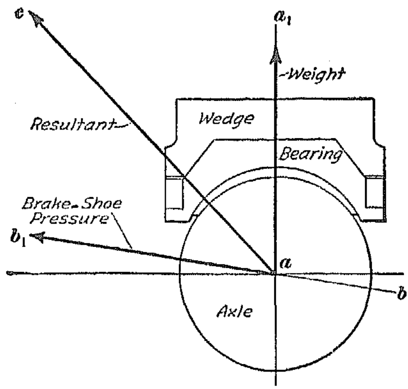
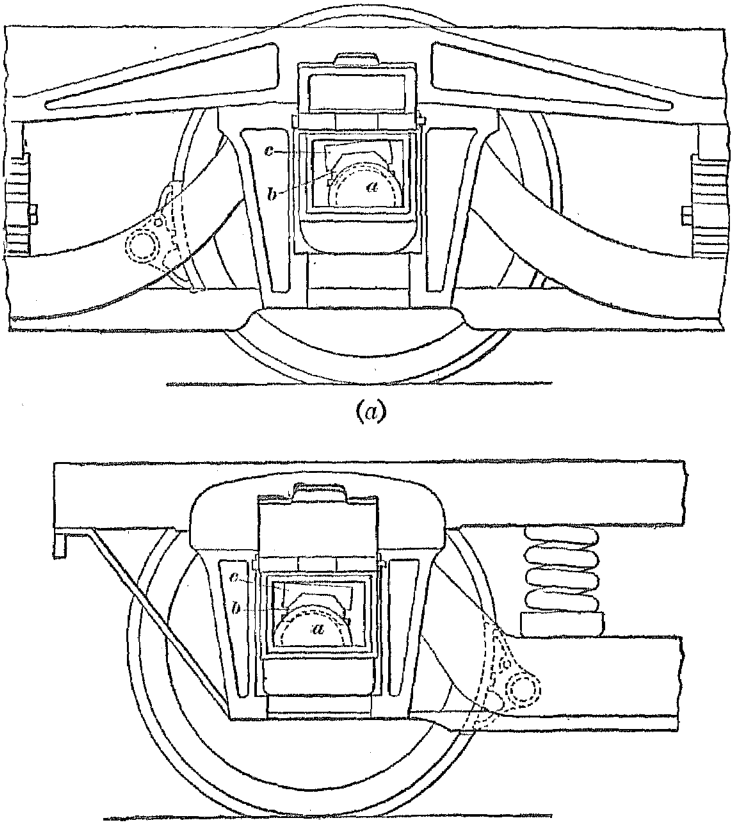


FIG. 12

10. Displacement of Journal.—When the brake shoes are hung low on the wheels, as they necessarily must be when a single brake shoe is used, there is danger of the axle journal being pushed out of its bearing when a heavy brake application is made. This can be understood by referring to Fig. 12, which illustrates a typical condition when one brake shoe is used. The downward force on the axle due to the weight is resisted by an equal and opposite upward force, which is represented by the line $a\ a_1$. The brake-shoe pressure tends to push the axle along the line $b\ b_1$ from the center of the shoe to the center of the wheel. The combined action of these forces causes the axle to be pushed by a force that acts in the direction of the line $a\ c$.

It will be noted that the line $a\ c$ passes near the outside corner of the wedge. Therefore, if the brake is applied hard the axle will tip up both the wedge and the bearing, and the axle will be forced against the journal box. This would not only cause the car to ride very roughly and the journal to overheat, but it would greatly increase the horizontal movement of the brake shoes and the false piston travel.

Views (*a*) and (*b*), Fig. 13, show the displacement of the journals, brasses, and wedges. The journal is marked *a*, the bearing *b*, and the wedge *c*. It will be noted that the journal is moved to the front of the box in Fig. 13 (*a*) and to the



(*a*)

(*b*)

Fig. 13

rear in (*b*). The brake shoe is therefore behind the wheel in Fig. 13 (*a*) and ahead of the wheel in Fig. 13 (*b*). When the journal bearing is displaced the journal pushes against the leg of the pedestal and is liable to break it.

11. Importance of Proper Design of Brake Rigging. The foregoing shows the importance of a design of brake rigging that will maintain the same piston travel under all conditions. A design which permits false travel gives rise to a great number of objectional features. For example, when the piston travel increases during an application of the brakes, the slack adjuster shortens the travel and reduces the shoe clearance. A short piston travel in turn destroys the flexibility of the brake, as light brake applications cannot be made without severe shocks.

12. Advantages of the Clasp Brake.—The principal advantage of a properly designed, manufactured, and installed type of brake rigging using two brake shoes per wheel is that there is practically no difference between the standing and the running piston travel. The reason is that the brake shoes can be located close to the horizontal center line of the wheels, because the braking forces on both sides of the wheel are balanced and there is no danger of displacing the journals. With the brake shoes hung in this manner no creeping action occurs as with the single shoe, and the brake shoes remain the same distance from the brake cylinder. The same piston travel under all conditions eliminates all the undesirable features that arise from a variable piston travel, such as reduced shoe clearance and loss of brake flexibility. Heavy overloads on the springs and equalizers and journal displacement with consequent overheating are also prevented.

An example of the superiority of the clasp-brake over the single-brake shoe in reducing false piston travel is shown in a test in which the increase in piston travel was only one-half inch from a service application of the brakes, standing, to an emergency application at 80 miles per hour. Cars equipped with a single-brake shoe under a similar test gave an increase in piston travel of 5 or 6 inches.

Another advantage of the clasp brake is the higher coefficient of friction and therefore the greater retarding force which is obtained when two brake shoes are used. The principal factor which affects the coefficient of friction under severe braking conditions is heat. When the braking force is applied to the

wheel through one brake shoe, the shoe must necessarily become much hotter than when the pressure is divided between two shoes. It has been found that the gain in the coefficient of friction with the clasp brake is over 5 per cent. The brake-shoe wear with two brake shoes will also be less than when one is used.

DIFFERENT DESIGNS OF BRAKE RIGGING

13. Conditions Affecting Design.—The design of the brake rigging as far as the arrangement at the brake cylinder is concerned, depends on whether the air-brake equipment consists of one or two brake cylinders. Two types of brake equipment are used on freight cars, the ordinary type of brake, also known as the single-capacity brake or empty-car brake, on account of the same braking force being used under all conditions of loading; and the empty and load brake, in which the breaking force varies. As the former uses one brake cylinder and the latter two, the design of the brake rigging necessarily differs.

The PM and LN equipments are one brake-cylinder equipments and the brake rigging, therefore, does not differ materially from that of the empty car brake. In cases where the weight of the car is such as to require a braking force greater than the capacity of one brake cylinder with the proper leverage ratio, two air-brake equipments known as the double PM and the double LN are used. The brake rigging for the double equipments consists of a separate set of brake rigging for each truck and does not differ from the rigging used with the single equipment. The PC equipment and some of the earlier UC equipments use two brake cylinders, both of which are connected to the same brake rigging, and the rigging, therefore, differs from that used with other brake equipments. The brake rigging for locomotives differs materially from that used on freight or passenger cars.

14. Brake Rigging for Freight Equipment.—The brake rigging used with a single-capacity freight brake has

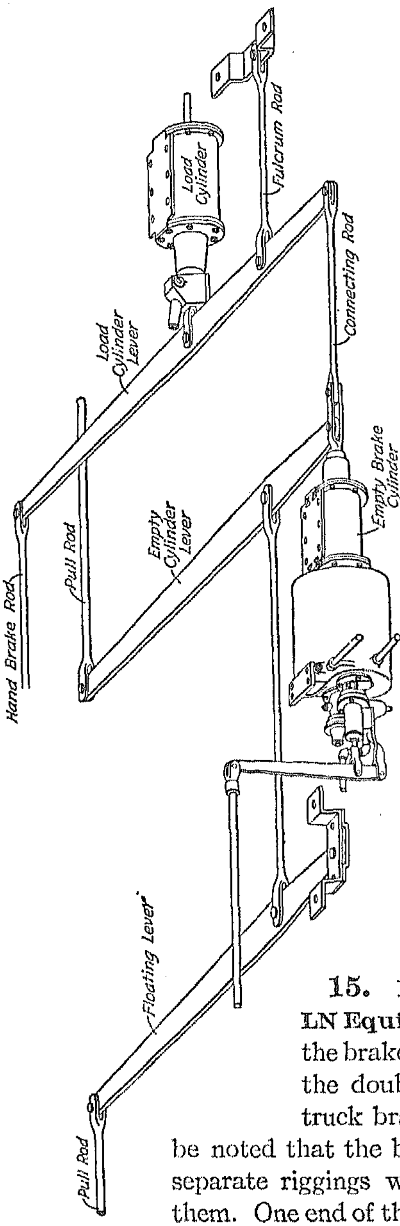


FIG. 14

already been illustrated and explained when considering Fig. 1. The brake rigging used with the empty and load brake is shown in Fig. 14. The brake rigging for the trucks is the same as is used with the single-capacity brake, and this part of the rigging is, therefore, omitted. The empty-cylinder lever is connected to the load-cylinder lever by a connecting-rod and the fulcrum rod forms a fulcrum for the latter lever. When the load cylinder is cut in, the force in this cylinder is transmitted to the push rod of the empty-brake cylinder and acts with the force that is being applied by the empty brake cylinder to the empty brake-cylinder lever.

15. Brake Rigging for Double LN Equipment.—In Fig. 15 is shown the brake rigging applied to a car with the double LN equipment with the truck brake rigging omitted. It will be noted that the brake rigging consists of two separate riggings with no connection between them. One end of the live cylinder lever works in

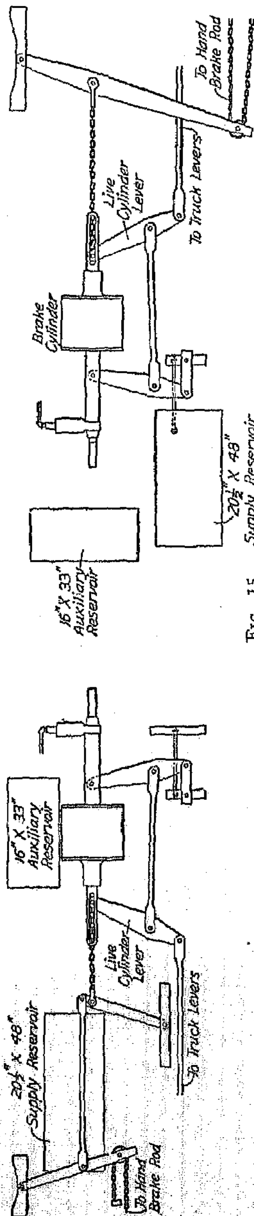


FIG. 15

a slot in the push rod, thereby enabling the hand-brake to be applied without moving the brake-cylinder piston.

16. Brake Rigging for PC Equipment.—In Fig. 16 is shown the part of the brake rigging adjacent to the brake cylinders, that is used with the PC equipment. With this equipment the emergency-brake cylinder is only used in an emergency application. The cylinder levers are attached to the pull rods by connecting links. The end of the cylinder lever which is connected to the crosshead of the service cylinder moves outwards when the brake is applied and draws each of the pull rods toward the brake cylinder. The end of the cylinder lever which is connected to the emergency cylinder moves outwards at this time in the slotted crosshead without moving the piston. When an emergency application is made all the cylinder levers transmit a pull to the pull rods, and the braking force is doubled, provided the brake cylinders are of the same size.

A hand-brake rod is connected to one end of each hand-brake lever and the lever is pinned to a fulcrum that is bolted to the car. The hand-brake operates in harmony with the air brake. This means that the application of the hand-brake operates the brake rigging in the same way as the air brake operates it. This is a legal requirement of the Interstate Commerce

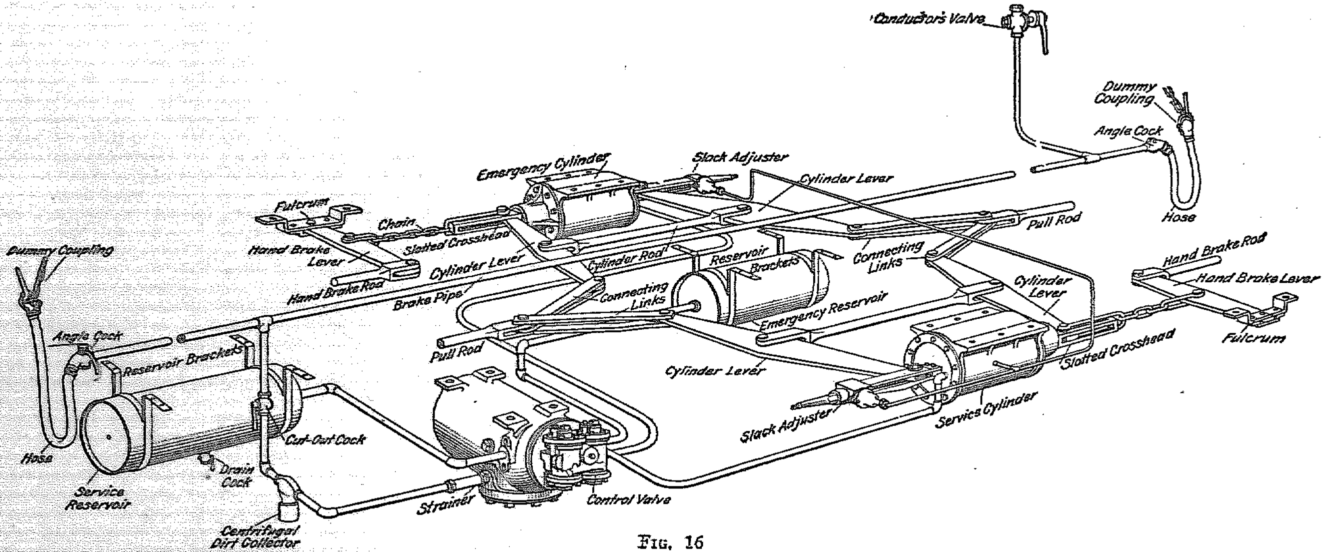

Conductor's Valve
Dummy Coupling
Angle Cock
Hose
Emergency Cylinder
Slack Adjuster
Cylinder Lever
Pull Rod
Fulcrum
Chain
Slotted Crosshead
Hand Brake Lever
Cylinder Rod
Reservoir Brackets
Connecting Links
Hand Brake Rod
Cylinder Lever
Hand Brake Lever
Hand Brake Rod
Brake Pipe
Connecting Links
Emergency Reservoir
Cylinder Lever
Fulcrum
Dummy Coupling
Angle Cock
Reservoir Brackets
Pull Rod
Slotted Crosshead
Cylinder Lever
Service Cylinder
Slack Adjuster
Cut-Out Cock
Hose
Drain Cock
Service Reservoir
Strainer
Control Valve
Centrifugal Dirt Collector

FIG. 16

Commission. The hand-brake does not operate in harmony with the air brake when the live cylinder lever moves toward the brake cylinder when the hand-brake is applied.

BRAKE-RIGGING CALCULATIONS

17. Reason for Figuring Leverage.—The principal reason for figuring leverage is to find the nominal braking ratio. It can then be seen whether the braking ratio is more or less than it should be for the service in which the car is engaged. When it is desired to find the braking ratio, one starts with a known size of brake cylinder and with a sketch of the brake rigging which shows the length of all the levers. The levers should be carefully measured, the measurements being taken between the centers of the pins or pin holes. The length of the lever arms is usually taken to the nearest $\frac{1}{4}$ inch. When figuring leverage one lever is taken at a time and, therefore, the rules which have already been given can be used for the calculation of a single lever.

As the pressure of the air in the brake cylinder is the force which is applied to the brake rigging through the brake-cylinder piston, the first step that must be taken when figuring leverage, is to find how to calculate the brake-cylinder force.

18. Force Exerted in Brake Cylinder.—The brake-cylinder force, or brake-cylinder value, as it is usually called, is equal to the area of the brake piston in square inches multiplied by the air pressure in pounds per square inch. The area of the brake piston is found by multiplying the diameter by the diameter, and then by .7854. If the diameter of a brake piston is 10 inches, the area is equal to $10 \times 10 \times .7854 = 78.5$ square inches.

Table I gives the brake-cylinder values for different pressures as well as the area of the brake-cylinder pistons of different diameters.

TABLE I

BRAKE-CYLINDER VALUES

Diameter	Area in Sq. In.	Force in Pounds			
		50 Lb. per Sq. In.	60 Lb. per Sq. In.	85 Lb. per Sq. In.	100 Lb. per Sq. In.
Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6
8″	50.1	2,500	3,000	4,250	5,010
10″	78.5	3,925	4,700	6,700	7,850
12″	113.1	5,650	6,800	9,600	11,300
14″	154.0	7,700	9,250	13,100	15,400
16″	201.0	10,050	12,050	17,100	20,100
18″	254.0	12,700	15,250	21,600	25,400
2– 8″	100.5	5,050	6,050	8,550	10,050
2–10″	157.1	7,850	9,400	13,400	15,710
2–12″	226.0	11,300	13,600	19,200	22,600
2–14″	308.0	15,400	18,500	26,200	30,800
2–16″	402.0	20,100	24,100	34,200	40,200
2–18″	508.0	25,400	30,500	43,200	50,800

The brake-cylinder value with a pressure of 86 pounds can be obtained by adding columns 2 and 5, because a pressure of 1 pound per square inch will give the figures shown in column 2.

FINDING THE BRAKING RATIO

19. Rules for Finding the Braking Ratio.—The rules by which all problems relating to levers can be solved were given in *Foundation Brake Rigging*, Part 1. It is only necessary, when it is required to find the braking ratio, to use two of these rules, which are repeated here for convenience.

Rule I.—*To obtain the force at a point between the ends of a lever, add together the forces on the ends. To obtain the force at one end, subtract the force at the other end from the force between the ends.*

Rule II.—*To find the delivered force, multiply the force applied by the length of the lever between the applied force and the fulcrum, and divide by the length of the lever between the delivered force and the fulcrum.*

20. Finding the Braking Ratio.—Suppose that it is required to find the braking ratio of a freight car weighing 52,000 pounds when empty. The braking ratio of such a car should be 60 per cent.; that is, 60 per cent. of the empty weight should be used as the brake-shoe pressure, or the braking force.

The levers of the foundation brake rigging are carefully measured and a sketch is made of the rigging, as shown in Fig. 17. The diameter of the brake cylinder is 10 inches, and the brake-cylinder force will be taken as 3,900 pounds instead of the calculated pressure of 3,925 pounds. When fractions occur in the calculations that follow, they are dropped, and whole numbers only are used. The points which are taken as fulcrums are marked *f*. The calculation of the braking ratio may be divided into successive steps, as follows:

21. Force Delivered to Top Rod by Live Cylinder Lever.—The force which is delivered to the top rod by the lower end of the live cylinder lever may be found by applying rule II, and it is, therefore, equal to

$$\frac{3{,}900 \times 12}{24} = 1{,}950 \text{ pounds}$$

22. Force Delivered to Brake Beam by Live Truck Lever.—A force of 1,950 pounds is applied by the top rod to the end of the live truck lever on the left truck. The force which the live truck lever delivers to the brake beam is equal to $\frac{1{,}950 \times 24}{6} = 7{,}800$ pounds. The braking force on each of the wheels is then equal to 3,900 pounds.

23. Force Delivered by Live Truck Lever to Bottom Rod.—The force delivered by the live-truck lever to the bottom rod is equal to $7{,}800 - 1{,}950 = 5{,}850$ pounds. This is the force which is applied to the lower end of the dead truck lever by the bottom rod.

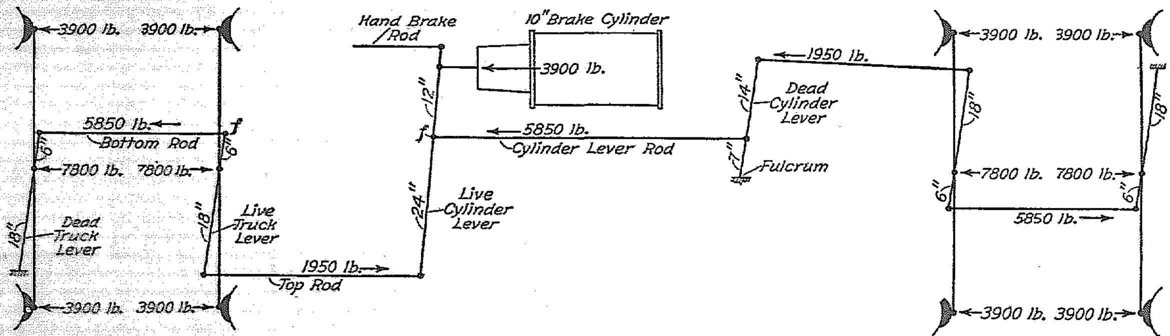

Hand Brake Rod
10" Brake Cylinder
3900 lb.
3900 lb.
3900 lb.
5850 lb.
Bottom Rod
7800 lb.
7800 lb.
Dead Truck Lever
Live Truck Lever
1950 lb.
Top Rod
12"
24"
Live Cylinder Lever
5850 lb.
Cylinder Lever Rod
1950 lb.
14"
Dead Cylinder Lever
7"
Fulcrum
18"
6"
7800 lb.
7800 lb.
5850 lb.

FIG. 17

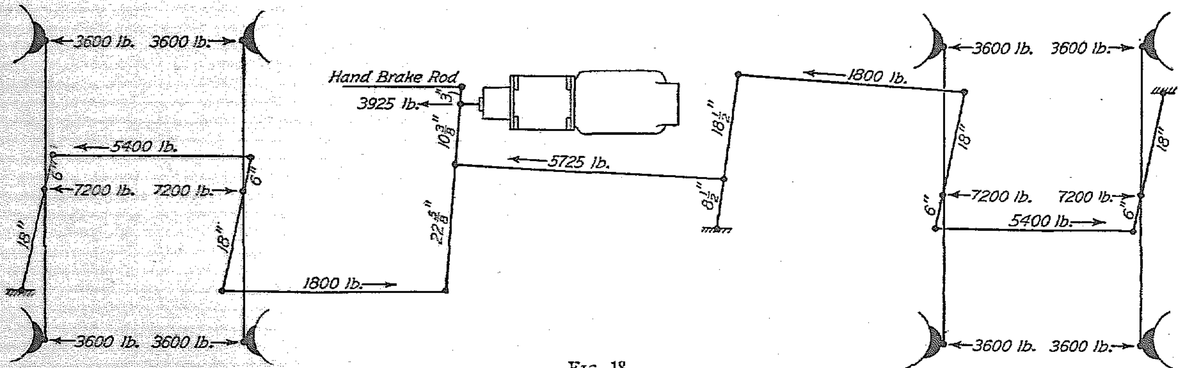

Hand Brake Rod
3925 lb.
3"
3600 lb.
3600 lb.
5400 lb.
7200 lb.
7200 lb.
6"
18"
1800 lb.
10 3/8"
22 5/8"
5725 lb.
18 1/2"
8 1/2"
1800 lb.
7200 lb.
7200 lb.
5400 lb.

FIG. 18

24. Force Delivered to Brake Beam by Dead Truck Lever.—The force delivered to the other brake beam by the dead truck lever, taking the upper end as the fulcrum, is

$$\frac{5{,}850\times 24}{18}=7{,}800 \text{ pounds}$$

The braking force on each of the wheels of this truck is then 3,900 pounds, or it is the same as that on the other truck

25. Force on Fulcrum of Live Cylinder Lever.—The force on the fulcrum of the live cylinder lever is equal to $3{,}900+1{,}950=5{,}850$ pounds. This is the force which is applied to the end of the dead cylinder lever through the cylinder lever rod.

26. Calculating the Forces on Other Truck.—The forces which act upon the brake beams of the right-hand truck are figured in the same way as on the left-hand truck, and the calculations, therefore, need not be repeated.

27. Total Brake-Shoe Pressure.—The total brake shoe pressure equals $7{,}800\times 4=31{,}200$ pounds

28. Braking Ratio.—The braking ratio is equal to the braking force (31,200 pounds) divided by the empty weight of the car (52,000 pounds) or .60, or 60 per cent. The braking ratio of the car is, therefore, correct.

29. Total Leverage Ratio.—The total leverage ratio is the total brake-shoe pressure (31,200 pounds) divided by the cylinder force (3,900 pounds). In this case the ratio is 8.

30. Fulcrum Points.—The fulcrum points which are taken are not the ones that must necessarily be used. For example, the fulcrum point of the live truck lever was assumed to be at f. However the point at which this lever is connected to the brake beam may also be taken as the fulcrum point. The application of rule II will then give the force which is delivered to the end of the lever. Thus,

$$\frac{1{,}950\times 18}{6}=5{,}850 \text{ pounds}$$

and from rule I, the force on the brake beam is

$$1{,}950+5{,}850=7{,}800 \text{ pounds}$$

Likewise the fulcrum of the dead cylinder lever may be assumed to be at the brake beam or at the upper end. If the fulcrum is taken at the brake beam, the force delivered to the upper end of the lever equals $\frac{5{,}850 \times 6}{18} = 1{,}950$ pounds. The force on the brake beam then is $1{,}950 + 5{,}850 = 7{,}800$ pounds. It may also be assumed that the fulcrum of the live cylinder lever is at the end instead of between the ends.

31. Locating the Middle Hole in the Live-Cylinder Lever.—Suppose it is required to find whether the middle hole in the live cylinder lever of a freight car is located at the correct point. The empty weight of the car is 48,000 pounds, the brake cylinder is 10 inches in diameter, and the measurement of the levers gives the lengths shown in Fig. 18.

32. Rules That Are Used.—The rules that are used when checking the length of the arms of the live-cylinder lever, are given in the previous Section but are repeated here for convenience.

Rule III.—*The applied force is equal to the delivered force multiplied by the length of the lever between the delivered force and the fulcrum, divided by the length of the lever between the applied force and the fulcrum.*

Rule IV.—*To find the length of the lever between the applied force and the fulcrum, multiply the delivered force by the total length of the lever, and divide by the sum of the applied and delivered forces. To find the length of the lever between the delivered force and the fulcrum, multiply the applied force by the length of the lever and divide by the sum of the applied and delivered forces.*

33. Order of Calculations.—The following order will be observed when the calculations which involve the location of the middle hole of the live-cylinder lever are taken up.

34. Total Braking Force or Brake-Shoe Pressure. The total braking force or brake-shoe pressure is equal to the empty weight of the car multiplied by the braking ratio. Thus, $48{,}000\times.60=28{,}800$ pounds.

35. Force on Brake Beams.—The force on each one of the brake beams is equal to $\frac{28{,}800}{4}=7{,}200$ pounds. This is the force which is delivered by the dead-truck lever.

36. Force Applied by Bottom Rod.—The force which must be applied to the dead-truck lever by the bottom rod to produce a force of 7,200 pounds on the brake beam may be found from rule III and is as follows:

$$\frac{7{,}200\times18}{24}=5{,}400 \text{ pounds}$$

This is the force delivered to the bottom rod by the live-truck lever.

37. Force Applied to Live-Truck Lever.—The force which is applied to the live-truck lever by the top rod is equal to $7{,}200-5{,}400=1{,}800$ pounds. This is the force which acts on the lower end of the live-cylinder lever.

38. Length of Arms of Live-Cylinder Lever.—The length of the arms of the live-cylinder lever or the location of the fulcrum point may be found from rule IV. The length of the lever arm between the applied force and the fulcrum is

$$\frac{1{,}800\times33}{1{,}800+3{,}925}=10\tfrac{3}{8} \text{ inches}$$

The length of the lever arm between the delivered force and the fulcrum is

$$\frac{3{,}925\times33}{1{,}800+3{,}925}=22\tfrac{5}{8} \text{ inches}$$

The live cylinder lever is, therefore, correctly proportioned.

CHECKING THE BRAKE RIGGING

39. The brake rigging, if properly designed, will transmit the same braking force to each wheel. It is then important to be able to check quickly the brake rigging to see whether it is properly designed. The brake rigging may be checked by obtaining the lengths of the lever arms between the centers of the pins or pinholes and then seeing whether the lever arms of all the truck levers and the cylinder levers are in the same proportion.

It will be noted that the term *lever arm* as here used means the length of the arms between the pinholes and does not mean the length of the arms between the applied and delivered forces and the fulcrum. Two levers are said to be in the same proportion if the same result is obtained when the length of the long arm of each is divided by the length of the short arm.

40. In Fig. 17, for example, the live- and the dead-cylinder levers are in the same proportion, because the same number, or 2, is obtained when the length of the long arm of each is divided by the length of the short arm. These levers are then said to be *2-to-1 levers*. All the truck levers are also in the same proportion, because each is a 3-to-1 lever. The brake rigging is, therefore, properly designed, because all levers of the same kind are in the same proportion and an equal braking force will be transmitted to each wheel.

It is not necessary for levers of the same kind to be of the same length. They may be of unequal lengths, provided the proportion remains the same. Thus, one truck lever may be 25 inches long and the other 30 inches long. The long arm of the first one is 20 inches and the short arm is 5 inches long. The long arm of the second lever is 24 inches and the short arm is 6 inches long. The levers are in the same proportion because each is a 4-to-1 lever.

41. Effect of an Improperly Proportioned Lever on Braking Force.—The effect on the braking force of substituting a lever which is not proportioned the same as the other

levers will now be explained. Let it be assumed, that one of the live truck levers in Fig. 17 is replaced by a lever 25 inches long, with one arm 20 inches long, and the short arm 5 inches long. This lever is then a 4-to-1 lever instead of a 3-to-1 lever, like the others. The following calculation will show the change that is made in the braking force by the substitution of this lever.

The force delivered to the first brake beam is

$$\frac{1{,}950\times 25}{5}=9{,}750 \text{ pounds}$$

The force which is delivered to the bottom rod is equal to $9{,}750-1{,}950=7{,}800$ pounds. The force which is delivered to the other brake beam is equal to $\frac{7{,}800\times 24}{18}=10{,}400$ pounds. The braking forces on the two pairs of wheels are unequal and the total braking force on the truck is $20{,}150-15{,}600$, or 4,550 pounds more than it should be.

However, a truck lever of a different length may be substituted if the proportion is the same as the old lever. Thus, a lever 28 inches long may be used if one arm is 21 inches long and the other arm is 7 inches long, because this is also a 3-to-1 lever. It is, therefore, necessary for all the truck levers and also the two cylinder levers to be in the same proportion.

42. Finding Braking Ratio With Clasp Brake.—In Fig. 8 (*b*) is shown the outline of a clasp brake applied to a six-wheel passenger-car truck. The brake-rigging arrangement is shown on only one side of the truck, as the arrangement is the same on the other side. The car weighs 147,860 pounds. The brake cylinder is 18 inches in diameter, and the brake-cylinder value is 15,250 pounds, with a pressure of 60 pounds. The lengths of the levers are as shown. The points taken as fulcrums are marked *f*.

The braking ratio can be found from rules I and II given in Art. **19**. When these rules are applied it will be noted that the values obtained in some cases are changed a few pounds so as

to have them all the same. It will be found convenient to arrange each step in the following order:

43. Force Delivered by Cylinder Lever to Horizontal Equalizer Connecting-Rod.—The force delivered by the lower end of the live cylinder lever a to the horizontal equalizer connecting-rod a_1 is

$$\frac{15{,}250\times 20\frac{1}{2}}{15\frac{1}{2}}=20{,}170 \text{ pounds}$$

This force is applied to the middle of the horizontal equalizer b and, therefore, a force of one-half this amount, or 10,085 pounds, is transmitted by this equalizer to the truck connecting-rod b_1 and the upper end of the first truck lever.

44. Force Delivered by Truck Lever to Hanger Lever. The force delivered by the truck lever through link d to the lower end of the hanger lever d_1 is

$$\frac{10{,}085\times 6\frac{1}{2}}{20\frac{3}{8}}=3{,}216 \text{ pounds}$$

45. Force Delivered to Brake Shoe.—The force delivered to the first brake shoe is

$$\frac{3{,}216\times 18\frac{3}{4}}{10\frac{7}{8}}=5{,}545 \text{ pounds}$$

46. Force Delivered to Second Truck Lever by Bottom Truck Rod.—The force delivered to the second truck lever by the bottom truck rod c is $10{,}085+3{,}216=13{,}300$ pounds.

47. Force Delivered to Hanger Lever by Truck Lever. The force delivered to the second hanger lever d_3 through the link d_2 is

$$\frac{13{,}300\times 6\frac{1}{2}}{26\frac{7}{8}\text{ (total length)}}=3{,}216 \text{ pounds}$$

48. Force Delivered to Brake Shoe.—The force delivered to the second brake shoe is

$$\frac{3{,}216\times 18\frac{3}{4}}{10\frac{7}{8}}=5{,}545 \text{ pounds}$$

49. Force Applied to Top Truck Rod.—The force applied to the top truck rod is 13,300−3,216=10,085 pounds.

50. Braking Force Delivered to Other Wheels.—The calculation of the braking force delivered to the other wheels is merely a repetition of the calculations already made, and it is therefore unnecessary to repeat them. The calculation of the braking force on the opposite side of the truck may be omitted for the same reason.

51. Calculating Braking Force on Other Truck.—It is unnecessary to calculate the braking force on the other truck, after it has been shown how to obtain the force which is delivered to the pull rod by the dead-cylinder lever. The force acting on the cylinder-lever connecting-rod is

$$20{,}170+15{,}250=35{,}420 \text{ pounds}$$

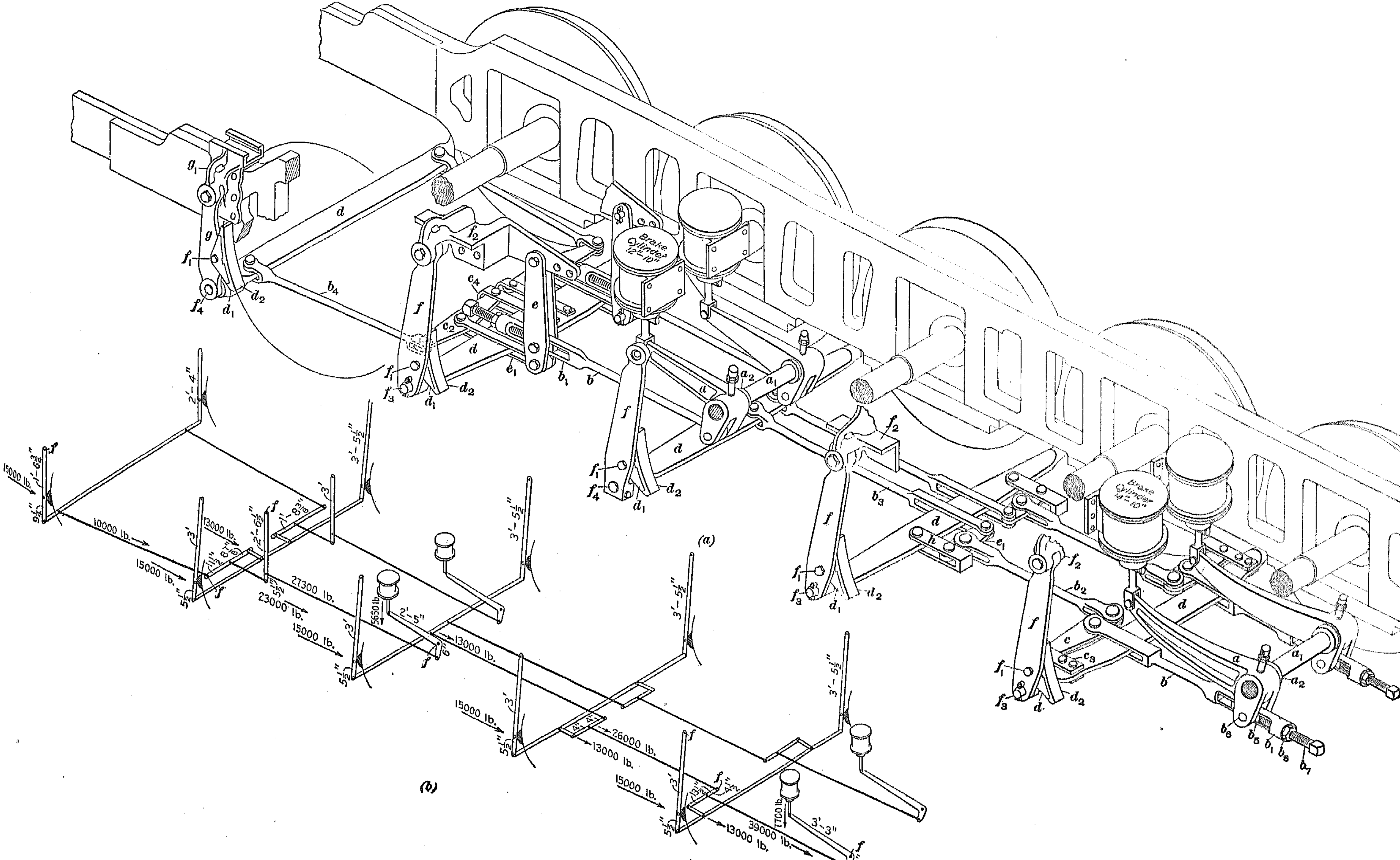
The force delivered by the lower end of the dead cylinder lever to the pull rod of the other truck is

$$\frac{35{,}420\times 20\tfrac{1}{2}}{36}=20{,}170 \text{ pounds}$$

52. Total Braking Force, or Brake-Shoe Pressure. The total braking force, or brake-shoe pressure, is equal to the force exerted against a wheel by one brake shoe (5,545 pounds) multiplied by the number of brake shoes (24). The total braking force is then about 133,080 pounds.

53. Service-Braking Ratio.—The service-braking ratio is equal to the service-braking force (133,080 pounds) divided by the weight of the car (147,860 pounds). The service braking ratio is then .90, or 90 per cent.

54. Emergency-Braking Ratio.—The emergency-braking ratio can be figured in the same way as the service-braking ratio. A brake-cylinder pressure of 100 pounds to the square inch, which is obtained in emergency with the LN and UC equipments, gives a brake-cylinder force of 25,400 pounds. (See Table I.) This force gives a braking force of about 221,760



Brake Cylinder 12"-10"
Brake Cylinder 14"-10"
(a)
(b)
15000 lb.
10000 lb.
13000 lb.
27300 lb.
23000 lb.
5650 lb.
2'-5"
26000 lb.
39000 lb.
7700 lb.
3'-3"
2'-4"
3'-5½"

Fig. 19

pounds when it is transmitted through the brake rigging to the brake shoes. The emergency-braking ratio is then

$$\frac{221,760}{147,860} = 1.5, \text{ or } 150 \text{ per cent.}$$

LOCOMOTIVE BRAKE RIGGING

55. Names of Parts.—In Fig. 19 (*a*) is shown the arrangement of the brake rigging as applied to a Santa Fe type of locomotive, and in view (*b*) is shown the length of the various levers and the stresses on the different points. In view (*a*) the frame of the locomotive on the right side is removed so as to show the rigging more clearly. Some parts of the brake rigging on the left side of the locomotive cannot be seen in this view, but this is unimportant, as these parts are the same as on the right side. Reference letters are applied to only one side of the brake rigging. The names of the principal parts of the brake rigging are as follows: a, bell crank; a_1, the shaft; b, pull rod with slack adjuster b_1; b_2, second pull rod; b_3, third pull rod; b_4, back pull rod; c, main equalizer; c_1, middle equalizer; c_2, back equalizer; d, brake beam; d_1, brake head; d_2, brake shoe; e, intermediate lever; e_1, pull connection; f, return hanger lever; g, hanger lever; h, strap link.

56. Arrangement of Parts.—The brake rigging shown in Fig. 19 (*a*) consists of two separate riggings, one operated by the two forward brake cylinders, and the other by the two rear brake cylinders. The reason for using four brake cylinders is that the braking force necessary on a heavy locomotive is more than two brake cylinders can develop. The two forward brake cylinders are 14 inches in diameter and 10 inches long. The brake-cylinder value is 7,700 pounds with a brake-cylinder pressure of 50 pounds to the square inch. The rear brake cylinders have a diameter of 12 inches and develop a force of 5,650 pounds with a pressure of 50 pounds to the square inch. The brake shoes hang behind the wheels so that there will be no pull downwards on the driver springs when the brakes are applied and the locomotive is moving ahead. The four bell-cranks a the back ends of which are pinned to the brake-cylinder

pistons, turn freely on their shafts a_1, which are connected to the frames of the engine by brackets not shown. The bellcranks are prevented from moving sidewise on the shafts by collars a_2, which are held in place by setscrews. The parts of the shafts on which the bellcranks turn are lubricated through the oil cups shown. The front pull rods b, which have a slack-adjuster arrangement b_1 where they connect to the bellcranks, are pinned to the main equalizers c, which in turn are pinned to a link c_3 on the brake beam. The second pull rods b_2 are pinned between the ends of the middle equalizers c_1. One end of these equalizers is pinned to the strap links h, which are pinned to the brake beam, and the other end is pinned to the third pull rod b_3. These pull rods are pinned directly to the third brake beam.

57. The front ends of the back pair of pull rods b are pinned to the bellcranks and their back ends are connected to the intermediate equalizers e by a slack-adjuster arrangement. The lower end of these equalizers are connected by the pull connections e_1 to the back equalizers c_2, which are connected to the brake beams by the links c_4. The outer ends of the back equalizers c_2 are pinned to the back pull rods b_4, which in turn are pinned to the back brake beam. The outer ends of the brake beams pass through holes in the return hanger levers f and the hanger levers g. The levers are held in place on the brake beams by cotter keys f_3 with the exception of the third and fifth brake beams, on which nuts f_4 are used. Collars are sometimes placed between the cotter keys and the levers to prevent any tendency for the keys to be sheared off. The brake shoes, with the exception of the back pair, are connected by pins f_1 to lugs cast on the inside of the return hanger levers. The upper ends of the hanger levers turn freely in the brackets f_2 and g_1, which are bolted to the frame of the engine.

Rod supports not shown in this view prevent any part of the brake rigging, if it should break, from falling on the track. Balance springs keep the brake shoes in their proper relation to the wheels.

58. The construction of the slack adjuster is as follows: The end of the front pull rod b_1 is slotted. A crosshead b_5 is

placed in this slot and is held in position by a pin b_6 which passes through the crosshead and both sides of the bellcrank. The inner end of the threaded adjuster rod b_7 bears against the front end of the crosshead. A jamb nut b_8 serves to keep the rod from turning. The arrangement for taking up the slack is the same on the rear pull rods as on the front rods. The slack is taken up by slacking off on the jamb nut and turning the adjuster rod to the right by applying a wrench to the square end. This draws the pull rod forward as the slot in it is longer than the crosshead. All the adjuster rods should be turned the same amount, after which all jamb nuts should be tightened.

59. Calculating the Braking Ratio.—Reference will be made to Fig. 19 (*b*) when calculating the braking ratio. The locomotive to which the brake rigging is applied has a weight of 300,000 pounds on the drivers, and it is required to find the braking ratio when the brake-cylinder pressure is 50 pounds per square inch.

First it is necessary to find the braking force or the shoe pressure, which can be done by applying the rules given in connection with the braking ratio of cars. It is necessary to calculate the braking force only on one side of the locomotive, as the braking force on the other side is the same. It will be found more convenient to reduce to inches the measurements which are shown in feet. The points on the different levers considered as fulcrums are marked *f*. It will be noted when calculating the forces that the results obtained do not agree exactly with the forces shown in view (*b*). However, the difference in all cases is so small that the odd numbers are dropped and whole numbers are used as shown in view (*b*). The dimensions on the hanger levers on the left side show their total length, and on the right side the length of these levers is shown between the pinholes.

60. The force applied to the main equalizer through the bellcrank and the pull rod is

$$\frac{7{,}700\times 39}{7\frac{11}{16}}=39{,}000 \text{ pounds}$$

The force applied to the brake beam is

$$\frac{39{,}000\times 4\frac{1}{2}}{13\frac{1}{2}}=13{,}000 \text{ pounds}$$

The force transmitted to the brake shoe is

$$\frac{13{,}000\times 41\frac{1}{2}}{36}=15{,}000 \text{ pounds}$$

The force applied to the middle equalizer is $39{,}000-13{,}000=26{,}000$ pounds, and as this force is applied to the middle of the equalizer, a force of 13,000 pounds will be applied to the brake beam. It is unnecessary to show how to obtain the brake-shoe pressure at this brake beam, as the same calculation is used as was used at the first brake beam. For this reason the calculations for the third and fourth brake shoes are also omitted.

The force delivered by the second brake cylinder through the bellcrank and pull rod to the intermediate equalizer is

$$\frac{5{,}650\times 29}{6}=27{,}300 \text{ pounds}$$

The force delivered by the intermediate equalizer to the back equalizer is

$$\frac{27{,}300\times 30\frac{1}{2}}{36}=23{,}000 \text{ pounds}$$

The force delivered to the brake beam is

$$\frac{23{,}000\times 11\frac{1}{2}}{20\frac{3}{8}}=13{,}000 \text{ pounds}$$

The force applied to the fourth brake shoe can be found in the same manner as for the first brake shoe, and is 15,000 pounds. The back pull rod transmits a force of $23{,}000-13{,}000=10{,}000$ pounds to the back brake beam. The force delivered to the back brake shoe is

$$\frac{10{,}000\times 28}{18\frac{3}{4}}=15{,}000 \text{ pounds}$$

61. **Braking Ratio.**—The total braking force is 15,000 × 10 = 150,000 pounds, and the weight on the drivers is 300,000 pounds. The braking ratio, therefore, is

$$\frac{150{,}000}{300{,}000} = .50, \text{ or 50 per cent.}$$

CALCULATIONS INVOLVING AIR PRESSURES

VOLUME OF CYLINDERS AND RESERVOIRS

62. **Volume of a Brake Cylinder.**—The volume of a brake cylinder may be found by applying the following rule:

Rule.—*To find the volume of a brake cylinder in cubic inches, multiply the cross-sectional area of the cylinder, in square inches, by the piston travel, in inches.*

EXAMPLE.—What is the volume of a 10-inch brake cylinder having an 8-inch piston travel?

SOLUTION.—The area of the brake cylinder is equal to the diameter multiplied by the diameter and by .7854, or to 78.5 sq. in. Hence, the volume of the brake cylinder is 78.5 × 8 = 628 cu. in. Ans.

The volume of a brake cylinder is greater than the amount calculated by the above rule, for the reason that there is an extra volume that the rule does not take into consideration. In freight equipment, the volume of the auxiliary tube and the cylinder clearance, which is the space between the brake-cylinder piston and the end of the auxiliary reservoir when the piston is in the position it assumes when the brake is released, must also be taken into account when calculating the volume of the brake cylinder. With passenger equipments the brake-cylinder clearance as well as the volume of the passage in the brake-cylinder head with some equipments or the volume of the piping between the universal valve and the brake cylinder with the UC equipment, must also be considered. Usually, about 48 cubic inches is added to the calculated volume of a brake cylinder to make up for cylinder clearance, etc.

In Table II the volumes of the standard brake cylinders are given, due allowance having been made for cylinder clearance, etc.

TABLE II

VOLUMES OF AIR-BRAKE CYLINDERS

Size of Cylinder Inches	Piston Travel Inches	Volume Cubic Inches
8	8	450
10	8	675
12	8	950
14	8	1,280
16	8	1,650
18	8	2,085

If the rule for the volume of a cylinder be applied for a piston travel of 1 inch, it will be found that the number of cubic inches the volume of a brake cylinder will change for each increase or decrease of piston travel, is numerically equal to the area of the cylinder. For example, the volume of an 8-inch cylinder will change $50\frac{1}{4}$ cubic inches for every 1 inch of change in the piston travel; that of a 10-inch cylinder will change 78.5 cubic inches; that of a 12-inch cylinder, 113 cubic inches; and so on.

VOLUME OF AUXILIARY RESERVOIR

63. Volume of Freight-Car Auxiliary Reservoir. The volume of the auxiliary reservoir used with an 8-inch brake cylinder is 1,620 cubic inches, and when a 10-inch brake cylinder is used, the volume of the auxiliary reservoir is 2,440 cubic inches. The auxiliary reservoirs used with freight equipment are cast iron, and the reservoirs used with passenger equipment are of riveted and welded construction.

64. Volumes of Auxiliary Reservoirs Used With Passenger-Car Equipments.—Tables III, IV, and V give

the volumes of the auxiliary reservoirs which are used with the different types of passenger equipment and also the sizes of the brake cylinders. The first number in the dimensions is the diameter and the second number is the length.

TABLE III

BRAKE CYLINDERS AND RESERVOIRS FOR PM AND LN PASSENGER-CAR BRAKE EQUIPMENT -- SINGLE BRAKE-CYLINDER INSTALLATION

Brake Cylinders	Auxiliary Reservoirs for Use With PM and LN Equipments		Supplementary Reservoirs for Use With LN Equipment Only	
Size Inches	Size Inches	Volume Cu. In.	Size Inches	Volume Cu. In.
10×12	12×27	2,450	16 ×33	5,724
12×12	12×33	3,088	16 ×48	8,577
14×12	14×33	4,476	20½×36	10,158
16×12	16×33	5,724	20½×48	14,003
18×12	16×42	7,436	22½×54	18,967

The same sizes of reservoirs and brake cylinders are used with the double brake-cylinder installation, or when each truck has a separate equipment. Such brake equipments are referred to as double PM or double LN equipments. The double brake equipments are used when the brake cylinder is 12×12 inches and upwards.

According to Table III, the auxiliary reservoirs and brake cylinders are so proportioned that with 70 pounds brake-pipe pressure there is obtained for service applications from a 20-pound reduction an equalized brake-cylinder pressure of 50 pounds, with an 8-inch piston travel.

TABLE IV

BRAKE CYLINDERS AND RESERVOIRS FOR PC PASSENGER CAR BRAKE EQUIPMENT

Brake Cylinders	Service Reservoirs		Emergency Reservoirs	
Size Inches	Size Inches	Volume Cu. In.	Size Inches	Volume Cu. In.
14×12	16 ×48	8,577	16 ×42	7,436
16×12	20½×36	10,158	16 ×48	8,577
18×12	20½×48	14,003	20½×36	10,158

The reservoirs and brake cylinders, according to Table IV, are so proportioned that with 100 pounds brake-pipe pressure there is obtained for service applications from a 24-pound reduction an equalized pressure of 86 pounds in one brake cylinder with 8-inch piston travel, and for an emergency application an equalized pressure of 86 pounds in both brake cylinders with a piston travel of 9 inches.

TABLE V

BRAKE CYLINDERS AND RESERVOIRS FOR UC PASSENGER-CAR BRAKE EQUIPMENT

Single Brake-Cylinder Installation

Brake Cylinders	Auxiliary Reservoirs		Service Reservoirs		Emergency Reservoirs	
Size Inches	Size Inches	Volume Cu. In.	Size Inches	Volume Cu. In.	Size Inches	Volume Cu. In.
14×12	10×33	2,125	12×33	3,088	18½×42	10,014
16×12	10×33	2,125	14×33	4,476	20½×42	12,083
18×12	10×33	2,125	16×33	5,724	22½×48	16,661

The reservoirs and brake cylinders used with the UC equipment, Table V, are so proportioned that with 110 pounds brake-pipe pressure there is obtained for service applications from a 24-pound reduction a pressure of 60 pounds with an 8-inch piston travel and for an emergency application an equalized pressure of 100 pounds with a 9-inch piston travel.

AIR-PRESSURE CALCULATIONS

EXPLANATION OF PRESSURES

65. Purpose of Calculations.—The purpose of air-pressure calculations is to determine the pressure that is obtained in the brake cylinder when air is discharged into it from the auxiliary reservoir. Before problems involving air pressures can be attempted it is necessary to have a knowledge of atmospheric pressure, absolute pressure, gauge pressure, and of the manner in which air is affected when it is expanded and compressed.

66. Atmospheric Pressure.—By *atmospheric pressure* is meant the pressure that the atmosphere exerts by its weight. The atmosphere consists of air which completely surrounds the earth and extends upwards about 15 miles. Although a cubic foot of air has but little weight, a column of air with an area of 1 square inch and a height of 15 miles exerts a pressure of 14.7 pounds at sea level. In all ordinary calculations, the weight of the atmosphere, or the atmospheric pressure, is assumed to be 15 pounds per square inch.

67. Measuring Air Pressure.—An air gauge registers pressure, because the pressure in the interior of a tube is greater than the pressure of the atmosphere that surrounds the tube. An air gauge, therefore, cannot be used to measure the pressure of the atmosphere, because the air exerts the same pressure on all parts of the gauge. It is then necessary to remove the atmospheric pressure from a part of the apparatus that is used to determine the pressure, before the atmospheric pressure can be determined.

68. Method of Measuring Atmospheric Pressure. A simple method of determining the atmospheric pressure is illustrated in Fig. 20 (*a*) and (*b*). The glass tube shown is about 4 feet long and is closed at the upper end. The inside diameter of the tube is $1\frac{1}{8}$ inches and it therefore has an area of about 1 square inch.

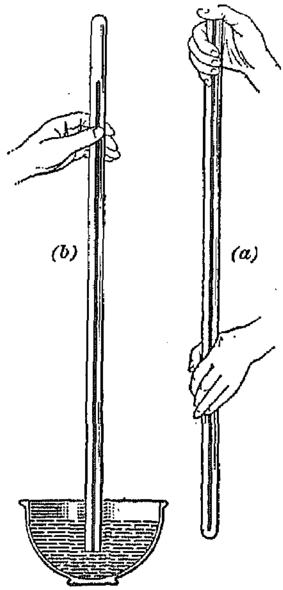


FIG. 20

The tube is filled with mercury, and the lower end is closed with the finger as shown in view (*a*). The tube is then inverted in the dish of mercury as shown in view (*b*) and the finger is removed. The weight of the atmosphere exerts a downward pressure on the mercury in the dish, and the pressure that is exerted on 1 square inch will now transmit an exactly equal upward pressure on the column of mercury in the tube, because the area of the column is also 1 square inch. The column of mercury will then descend until the weight of the mercury becomes the same as the upward pressure exerted against it by the weight of the air. The weight of the mercury which then remains in the tube must be equal to the pressure of the atmosphere per square inch. The mercury will be found to weigh about 15 pounds, which is, therefore, the pressure that the atmosphere exerts on 1 square inch on account of its weight.

69. Absolute Pressure and Gauge Pressure.—Absolute pressure is reckoned from the point of absolutely no pressure, and gauge pressure is reckoned from the point of atmospheric pressure, which is 15 pounds. The following explanation will serve to show the difference between absolute pressure and gauge pressure:

If a reservoir contains no air and it has attached to it a gauge

and also some device that will register atmospheric pressure and air is admitted to the reservoir, the device will show an increase in pressure, but the gauge hand will not move. When the reservoir is charged to atmospheric pressure the device will show 15 pounds pressure. If compressed air is now allowed to enter the reservoir, the gauge will immediately begin to register an increase of pressure, as well as the device that heretofore was registering atmospheric pressure. The device will then show 15 pounds more pressure than the gauge and this difference will be maintained as the pressure is increased. The pressure shown on the device is known as *absolute pressure*, and the pressure on the gauge as *gauge pressure*. To convert absolute pressure to gauge pressure, it is, therefore, merely necessary to subtract 15 pounds from the absolute pressure. To convert gauge pressure to absolute pressure, add 15 pounds to the gauge pressure. The gauge pressure may be considered as the pressure which tends to burst a reservoir. The first 15 pounds of absolute pressure exerts no pressure on the interior of a reservoir, because it is balanced by an equal atmospheric pressure on its exterior.

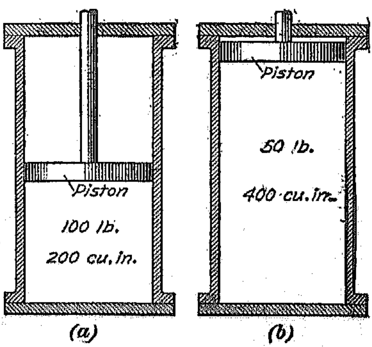


FIG. 21

VOLUME AND PRESSURE

70. Relation.—A certain definite relation exists between the volume of a certain quantity of air and the pressure to which the air is subjected, provided the temperature of the air remains the same. Reference will be made to Fig. 21 when explaining the relation between volume and pressure. The cylinders shown in Fig. 21, views (*a*) and (*b*), are of the same volume, and the pistons in them are assumed to work air-tight. The cylinder shown in view (*a*) contains air at an absolute pressure of 100 pounds to the square inch and the air is compressed into a space of 200 cubic inches.

Just how compressed air is affected when it is expanded will be shown first. If the force that acts downwards upon the piston in view (*a*) is gradually reduced, the pressure of the air will move the piston upwards, and view (*b*) shows that the pressure will decrease to 50 pounds to the square inch after the piston has moved, until the volume of the cylinder has become 400 cubic inches. Therefore, the pressure is decreased by one-half when the volume of the air is doubled.

How air is affected when it is compressed will next be shown. If the force which acts upon the piston in view (*b*) is increased until the air is compressed to a pressure of 100 pounds to the square inch, the air will occupy a volume of 200 cubic inches, as in view (*a*). The pressure is, therefore, doubled when the volume is decreased by one-half.

71. Rules Which Can Be Deduced.—It will be noted when the air is expanded, view (*a*), that the original pressure of the air multiplied by the volume of the air before it is expanded is equal, view (*b*), to the pressure of the air after it has been expanded, multiplied by the volume of the air after it has been expanded. Thus, $100 \times 200 = 50 \times 400$. It will also be noted when the air is compressed, view (*b*), that its pressure before compression multiplied by its volume before compression, is equal to the pressure after compression, multiplied by its volume after compression. Thus $50 \times 400 = 100 \times 200$. The following general rule can be deduced from the foregoing:

Rule I.—*The absolute pressure multiplied by the volume of the air before it is expanded or compressed is equal to the absolute pressure multiplied by the volume of the air after it is expanded or compressed.*

From this general rule three rules may be deduced and by means of these rules all problems which relate to brake-cylinder pressure may be solved. These rules as well as examples which will show how the rules are applied, will now be given.

Rule II.—*To find the absolute pressure after air is expanded, multiply the pressure before expansion by the volume before expansion and divide by the volume after expansion.*

EXAMPLE.—A reservoir with a volume of 540 cubic inches is charged with air at an absolute pressure of 150 pounds. The reservoir is connected to another reservoir which has a volume of 360 cubic inches. What is the absolute pressure and the gauge pressure when the pressure in both reservoirs has become equal?

SOLUTION.—The volume occupied by the air before expansion is 540 cubic inches, and the volume which the air occupies after expansion is $540 + 360 = 900$ cu. in. Therefore, from rule II, the absolute pressure after expansion is

$$\frac{150 \times 540}{900} = 90 \text{ lb. absolute pressure}$$

or $90 - 15 = 75$ lb. gauge pressure. Ans.

Rule III.—*To find the volume occupied by the air after it has been expanded, multiply the pressure before expansion by the volume before expansion, and divide by the pressure after expansion.*

EXAMPLE.—With the same reservoir and the same pressure as given in the example under rule II, find the volume occupied by the air after the pressure has been reduced to 1 pound absolute pressure.

SOLUTION.—From rule III, the volume occupied by the air is

$$\frac{150 \times 540}{1} = 81{,}000 \text{ cu. in.} \quad \text{Ans.}$$

It will be noted that the last example is solved by merely multiplying the pressure by the volume.

Rule IV.—*To find the pressure which results after air has been compressed, multiply the volume of the air before compression by the pressure before compression, and divide by the volume after compression.*

EXAMPLE.—What is the pressure when 500 cubic inches of air at 1 pound absolute pressure has been compressed into a space of 25 cubic inches?

SOLUTION.—From rule IV, the pressure is

$$\frac{500 \times 1}{25} = 20 \text{ lb., absolute pressure, or 5 pounds gauge pressure}$$

72. When Gauge Pressure May Be Used.—Gauge pressure may be used when the space into which the air expands already contains air at atmospheric pressure or at more than atmospheric pressure. Absolute pressure must always be used when air is expanded into a vacuum, which is a space which contains no air pressure. The space formed in the brake cylinder when the brake piston moves out contains practically no air pressure. Therefore, absolute pressure must be used when it is required to calculate the brake-cylinder pressure developed from a brake application.

APPLICATION OF AIR-PRESSURE RULES

BRAKE-CYLINDER PRESSURES

73. Calculating Pressure of Equalization.—The pressure obtained when the air in the auxiliary reservoir is allowed to enter the brake cylinder until the pressures are equal may be found by applying rule II of Art. 71.

The solution of the following example shows how this rule is applied:

Example.—The auxiliary reservoir of a standard 10-inch freight equipment is charged to a pressure of 70 pounds to the square inch. What is the pressure of equalization when the piston travel is 8 inches?

Solution.—The volume of the auxiliary reservoir, and, therefore, of the air before expansion, is 2,440 cu. in. The absolute pressure before expansion is 85 lb. The volume of the brake cylinder is 675 cu. in. The volume of the air after expansion is equal to the combined volume of the auxiliary reservoir and the brake cylinder, or 2,440 + 675 = 3,115 cu. in. From rule II, the gauge pressure of equalization is

$$\frac{85 \times 2{,}440}{3{,}115} - 15 = 51.5 \text{ pounds}$$

The calculated brake-pipe reduction necessary to cause equalization is, then, 70 − 51.5 = 18.5 pounds and is not equal to 20 pounds, although the latter figure is the one that is always used.

CALCULATING LENGTH OF A STOP

74. Length of Stop on Level Track.—The distance in which a passenger train can be stopped on straight level track can be calculated from the following formula:

$$S = 1.467Vt + \frac{V^2}{30Pef}$$

in which S = length of stop, in feet; from time brakes start applying;

V = speed, in miles per hour;

P = nominal or calculated braking ratio, (expressed as a decimal, not as per cent.);

ef = overall efficiency of foundation brake rigging (e), multiplied by the average coefficient of brake-shoe friction (f);

t = time for brakes to apply, in seconds; this ranges from .75 second for electro-pneumatic brake, 12 cars, to 2.5 seconds for pneumatic brake, 12 cars, depending upon the type of air-brake equipment and length of train.

The reason for the number 1.467 in the formula is that its use converts miles per hour into feet per second. There are 5,280 feet in a mile and 3,600 seconds in an hour, so at one mile per hour the speed in feet per second is $\frac{5,280}{3,600}$ or 1.467. At 10 miles per hour the speed, in feet per second, is 10×1.467, or 14.6.

The values of e multiplied by f for various speeds, kinds of brake rigging, and types of brake shoes are given in Table VI.

EXAMPLE.—Find the distance in which a train can be stopped on straight level track, moving at a speed of 60 miles per hour, with clasp brake, plain brake shoes, and braking ratio of 150 per cent. (1.5), if it is assumed that it takes 3 seconds for the brakes to apply.

SOLUTION.—From the formula, the length of the stop is equal to

$$1.467 \times 60 \times 3 + \frac{60 \times 60}{30 \times 1.5 \times .094} = 1,115 \text{ feet. Ans.}$$

TABLE VI

EFFICIENCY FACTORS (VALUES OF $e \times f$)

Kind of Brake Rigging		Clasp Brake		Single Shoe*	
Type of Brake Shoe		Plain	Flanged	Plain	Flanged
Speed m.p.h.	Per Cent. Braking Ratio				
30	125=1.25	0.141	0.169	0.108	0.112
	150=1.5	0.129	0.154	0.099	0.103
	180=1.8	0.118	0.141	0.090	0.094
60	125=1.25	0.103	0.122	0.074	0.090
	150=1.5	0.094	0.112	0.068	0.082
	180=1.8	0.086	0.102	0.062	0.075
80	125=1.25	0.092	0.109	0.070	0.074
	150=1.5	0.084	0.100	0.064	0.068
	180=1.8	0.077	0.092	0.059	0.062

*Value of data uncertain due to non-uniform brake-shoe conditions.

75. Length of Stop on Descending Grades.—The distance in which a train can be stopped on a descending grade can be found from the following formula:

$$S = 1.467Vt + \frac{V^2}{30(Pef - G)}$$

in which G = the grade expressed as a decimal; the other letters have the same meaning as in the other formula.

EXAMPLE.—Find the distance in which a train can be stopped on a 1 per cent. descending grade, moving at a speed of 30 miles per hour, with clasp brake, plain brake shoes, and a braking ratio of 125 per cent., if it is assumed that it takes 2.5 seconds for the brakes to apply.

SOLUTION.—From the formula the length of the stop is equal to

$$1.467 \times 30 \times 2.5 + \frac{30 \times 30}{30(1.25 \times .141 - .01)} = 290.5 \text{ ft.} \quad \text{Ans.}$$

76. Length of Stop on Ascending Grades.—The distance in which a train can be stopped on an ascending grade can be found from the formula in Art. 75 by changing the minus sign before G to a plus sign.

77. A study of Table VI will show that the data given apply only to passenger-car trains, because the braking ratios of 125, 150, and 180 per cent., are only used on these trains. No data have been compiled on the value of e and f for the lower braking ratios of cars used in freight service, although 10 per cent. may be taken as an approximation.

It is interesting to note, in Table VI, which was compiled from experimental data, how the coefficient of brake-shoe friction decreases as the speed and the braking ratios are increased. Thus, at 30 miles per hour and a braking ratio of 125 per cent., the coefficient of friction is .141, or 14.1 per cent. To convert a decimal to per cent. multiply by 100.

At 60 miles per hour and the same braking ratio, the coefficient of friction is .103, or 10.3 per cent., and at 80 miles per hour it is 9.2 per cent. The frictional coefficient also decreases as the braking ratio is increased, with the speed the same. Thus, at 30 miles per hour and a braking ratio of 125 per cent., the coefficient of friction is 14.1 per cent., whereas it is .118, or 11.8 per cent., with a braking ratio of 180 per cent.

The decrease in the coefficient of friction with the speed is due to the greater heat generated between the brake shoes and the wheels, the friction of the brake shoes reducing as the heat of the metal increases.

HAND BRAKES

DESCRIPTION AND CALCULATION OF FORCE

78. ARA Hand Brakes.—The hand-brake arrangements shown in Figs. 22, 23, and 24, together with the data given, have been reproduced from the ARA *Manual of Standard and Recommended Practice*, Section E. In this manual is the following statement: "The hand-brake wheel, or hand-brake ratchet lever, the brake staff at the chain, and the hand-brake

$$W=2\left(\frac{125\times L\times A}{r\times B}\right)$$

$$S=2\left(\frac{12\times A}{B}\right)$$

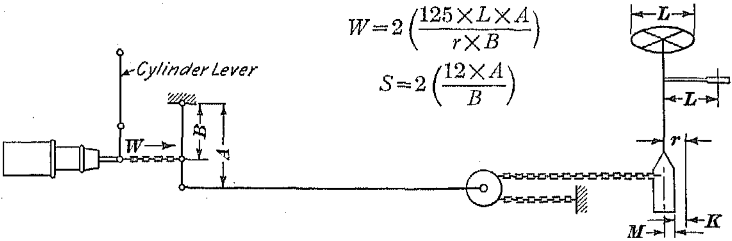


FIG. 22

$$W=\frac{125\times L\times A}{r\times B}$$

$$S=\frac{12\times A}{B}$$

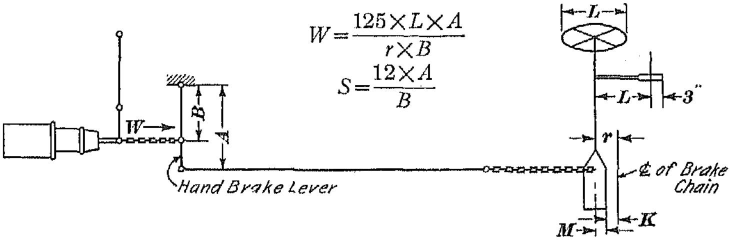


FIG. 23

$$W=\frac{2\times 125\times L}{r}$$

$$S=2\times 12=24$$

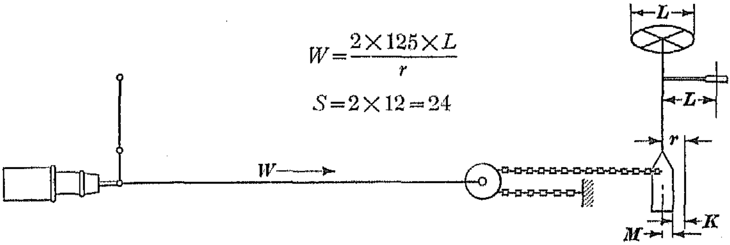


FIG. 24

W = equivalent pull, in lbs., at brake-cylinder piston rod;
125 = assumed pull, in lbs., on rim of brake wheel or on hand-brake lever 3 inches from outer end;
L, r, A, and B = dimensions, in inches;
$r=M+K$;
M = radius, in inches, of brake-shaft drum;
K = distance, in inches, from face of brake-shaft drum to center line of brake chain; for $\frac{7}{16}$-inch brake chain, $K=\frac{1}{2}$ inch;
S = minimum slack, in inches, to be taken up on brake staff;
12 = piston travel, in inches.

NOTE.—Dimension B not to be less than 11 inches.

leverage between the brake staff and the cylinder shall be so proportioned that a force of 125 pounds at the rim of the brake wheel or at 3 inches from the outer end of the hand-brake ratchet lever will develop an equivalent load at the brake-cylinder piston of not less than 2,500 and 3,95) lb., respectively, for cars having 8-inch and 10-inch cylinders."

The diameter of the hand-brake wheel recommended by the ARA is 16 inches and it is assumed that a man when operating the brake wheel with both hands can exert a force of 125 pounds on each side of the wheel. Therefore, two forces of

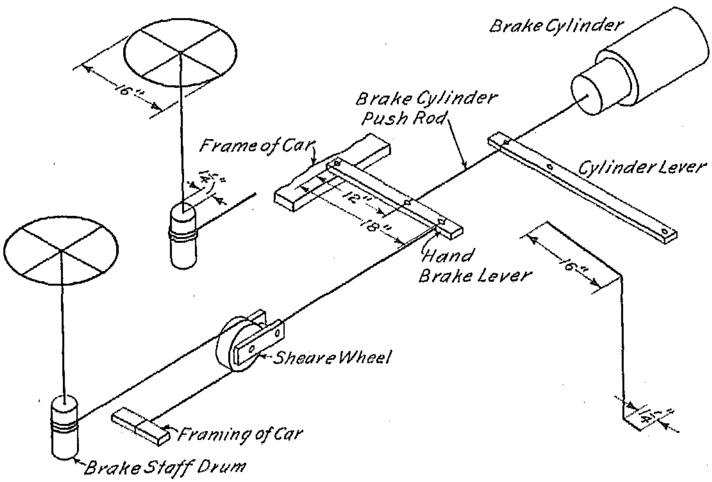


Fig. 25

125 pounds may be considered as acting through two separate levers 8 inches long as with the brake wheel, or a force of 125 pounds as acting through a lever 16 inches long as with the hand-brake ratchet lever.

79. The following explanation will show how the formula given in Fig. 22 is derived. In Fig. 25, the diameter of the brake wheel or the effective length of the hand-brake ratchet lever is assumed to be 16 inches, and the radius of the brake-shaft drum is assumed to be $\frac{3}{4}$ inch. Also, it will be assumed that a $\frac{7}{16}$-inch brake chain is used, which, it

will be noted, adds $\frac{1}{2}$ inch to the radius of the brake drum, so that the distance between the center of the drum and the brake chain, will be $\frac{3}{4}$ inch plus $\frac{1}{2}$ inch, or $1\frac{1}{4}$ inches. Then the force transmitted to the brake chain adjacent to the drum will be

$$\frac{125\times16}{1\frac{1}{4}}=1{,}600 \text{ lb.}$$

The 1,600 pounds transmitted to the brake chain is doubled by the action of the sheave wheel to 3,200 pounds, which will be the force that acts at the end of the hand-brake lever. The total length of this lever is 18 inches and the other dimension is 12 inches, hence the pull on the brake-cylinder push rod will be

$$\frac{3{,}200\times18}{12}=4{,}800 \text{ lb.}$$

This force exceeds the 3,950 pounds required, but the excess will take care of the losses due to friction and to other causes. The foregoing reasoning can be summarized in the formula given in Fig. 22. Substituting the values already assumed in this formula, it becomes

$$2\left(\frac{125\times16\times18}{12\times1\frac{1}{4}}\right)=4{,}800 \text{ lb.}$$

It is unnecessary to derive the formula shown in Figs. 23 and 24, as this can be easily done from the explanation already given.

80. Minimum Slack.—The letter S in Figs. 22, 23, and 24 refers to the length of chain, in inches, that must be wound up on the brake staff in order to apply the brake with the piston travel at its maximum, or 12 inches. This is known as minimum chain slack. The minimum amount of slack necessary to accommodate the full travel of the piston can be found by solving for S in the formula given for each figure. Because of the ratio of the hand-brake lever, Fig. 23, an amount of chain greater than the maximum piston travel must be wound up on the drum in order to draw the push rod out of the cylinder to the point where the brake shoes contact the wheels. In Fig. 24 the amount will be double because of the sheave

wheel, which makes it necessary to wind up a length of chain double the piston travel before the brake shoes make contact. Further complications are introduced in Fig. 22, which has both a lever and a sheave wheel, but if the proper formula is used in each case the minimum amount of slack can be readily calculated.

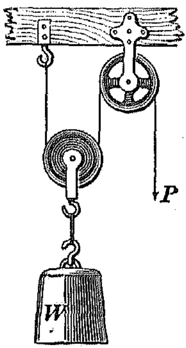


FIG. 26

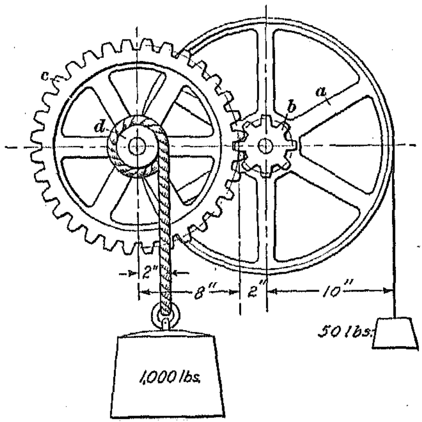


FIG. 27

In addition to the length of chain that winds up on the brake staff, a small additional amount is added to permit enough sag in the chain for the push rod to return to release position. Slack in excess of this is undesirable, as it means that more chain will have to be wound up on the brake staff.

POWER HAND BRAKES

81. Difference Between ARA Hand Brake and Power Hand Brake.—When considering the difference that exists between the ARA type of hand brake, and a power hand brake reference will first be made to Fig. 26. In this illustration, a movable pulley is shown connected to a weight, and a rope passes under this pulley and over the fixed pulley that is connected to the beam. With this arrangement a force of 50 pounds at P will balance a weight of 100 pounds. The

explanation is that the weight is supported by the parts of the cord on each side of the movable pulley, and as these parts are equally stretched, the weight must be equally divided between them. In other words, the point of support of the rope at the beam sustains one-half of the weight, and the point of support of the fixed pulley the other half. But the end of the cord at P must move through 2 feet to raise the weight 1 foot.

The arrangement shown in Fig. 27, which comprises a wheel a on which is fixed a pinion b that meshes with a gear c, will be considered next. The drum d on the gear c has connected to it a rope that carries the weight shown. Let it be assumed that the radius of the wheel a is 10 inches and the radius of the pinion b is 2 inches and that a force of 50 pounds is applied at the rim of the wheel. Then, according to the law of levers, the force applied to a tooth of the gear c will be $\frac{50 \times 10}{2}$, or 250 pounds. Now, if the radius of the gear c is 8 inches and the radius of the drum d is 2 inches, the pull of the rope is $\frac{250 \times 8}{2}$, or 1,000 pounds. It will be noted that the arrangement is merely a combination of levers but, unlike the ordinary bar lever, the movement of the ends of which is restricted, the gears comprise a set of levers of unlimited movement, or they may be called a continuous lever.

82. The important difference between the device shown in Fig. 26 and the one shown in Fig. 27 is that, with the gear and pinion arrangement, the force is multiplied before it is transmitted to the rope, whereas with the other arrangement with its movable pulley or sheave wheel the multiplication of the force occurs between the ends of the rope. The effect of stepping up the force before it is delivered to the end of the rope is to cause it to move an inch for each inch it is turned around the drum. On the other hand, the effect of increasing the force at a point between the ends of the rope is to cause a great deal of the rope to be pulled out in order to raise the weight. Hence, if the power point, Fig. 26, is taken as repre-

senting the point of attachment to the brake drum of the ARA hand brake and the movable pulley as the sheave wheel, a considerable amount of chain must be wound up before the brake shoes make contact with the wheels. Thus, in Fig. 24 the minimum amount of slack to be taken up at the brake staff will be 24 inches, or twice the maximum piston travel. As

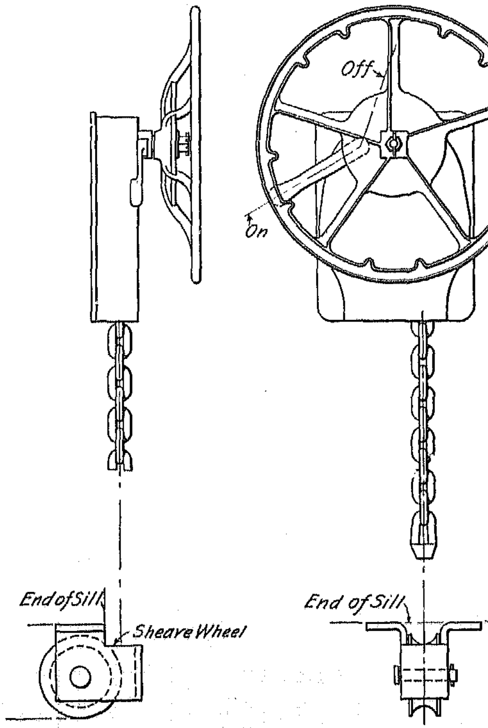


Fig. 28

the chain usually winds upon itself, the effect is to increase the diameter of the shaft and thereby reduce the delivered force. On the other hand, with a power hand brake, which employs a gear and a pinion, all the slack that has to be taken up will be an amount corresponding to the piston travel plus the sag, and this can be easily accommodated by a partial turn of the chain on the drum. Thus, 12 inches of chain would only make about one turn on a 4-inch drum.

With the same force developed by both types of brakes, the same number of revolutions would have to be made by both hand wheels to bring the brake shoes against the wheels, but as the hand wheel of the power brake can be spun, this brake can be applied more rapidly.

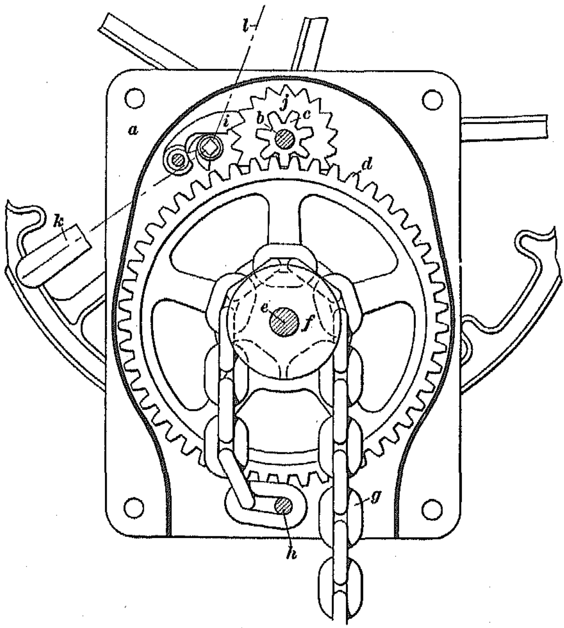


FIG. 29

83. Peacock Hand Brake.—The Peacock hand brake, Fig. 28, is attached in an approved location as one unit at the end of the car. In order to simplify the arrangement and make its operation clear in one view, the hand wheel with its shaft, pinion, and ratchet is shown applied in Fig. 29 from the rear instead of from the front as it actually is. The housing *a* contains a shaft *b*, one end of which extends through the casing, and on which the hand wheel shown in outline is mounted. On the same shaft is mounted a pinion *c* with seven teeth, and this pinion meshes with a gear *d* with 49 teeth. Cast integral with the gear *d*, which is mounted on a shaft *e*, is a chain drum *f* so constructed that the links of the chain *g*

wind over the drum and not around it. One end of the chain is pinned at h and the long end connects to a rod that extends down almost to the end sill. The vertical movement of the rod is converted into a horizontal movement by a malleable iron sheave located at the bottom of the end sill.

When the hand brake is applied there is a tendency for it to unwind, but this action is prevented by a pawl i that meshes with the ratchet wheel j, when the trip lever k is in its downward position. The brake can be released rapidly by moving the trip lever k to its upward position because this disengages the pawl i from the ratchet wheel j. The ratchet wheel j is mounted securely to the hand wheel shaft b.

84. Calculating the Force.—The force developed by the Peacock brake can be calculated from the following:

$$W = \frac{125 \times 24 \times 49}{7 \times 2.25} = 9{,}333 \text{ lb.}$$

In this formula, 125 = assumed pull, in pounds, on rim of brake wheel;
24 = diameter of hand wheel, in inches (corresponds to L);
49 = number of teeth in chain drum gear;
7 = number of teeth in pinion gear;
7 = gear ratio ($\frac{49}{7} = 7$);
2.25 = radius of chain drum (corresponds to r).

85. Ajax Hand Brake.—The arrangement of the Ajax hand brake applied to the end of a car is shown in Fig. 30 (a), and in (b) is shown a view with the back cover-plate removed. In this view, some of the parts are reversed in order to make them visible. The parts comprise a housing a, a hand-wheel shaft b, a pinion c with a pitch diameter of 2 inches that meshes with a gear wheel d. Cast with the gear wheel, which has a pitch diameter of $11\frac{1}{2}$ inches, is a chain drum e with a radius of 2 inches. The gear wheel and the chain drum are carried on a shaft that is supported at the ends in the housing. The chain f is pinned at the lower end to a vertical hand-brake rod g, the lower end of which is pinned to a bell-crank h, pivoted at i to the end sill by a bracket. The horizontal hand brake

rod *j* is connected by a chain *k* to the bell-crank; the other end is secured to the push rod or to the live cylinder lever by a short chain. The hand-wheel shaft also supports a ratchet wheel *l* controlled by a pawl *m*, operated by the lever *n*. When the

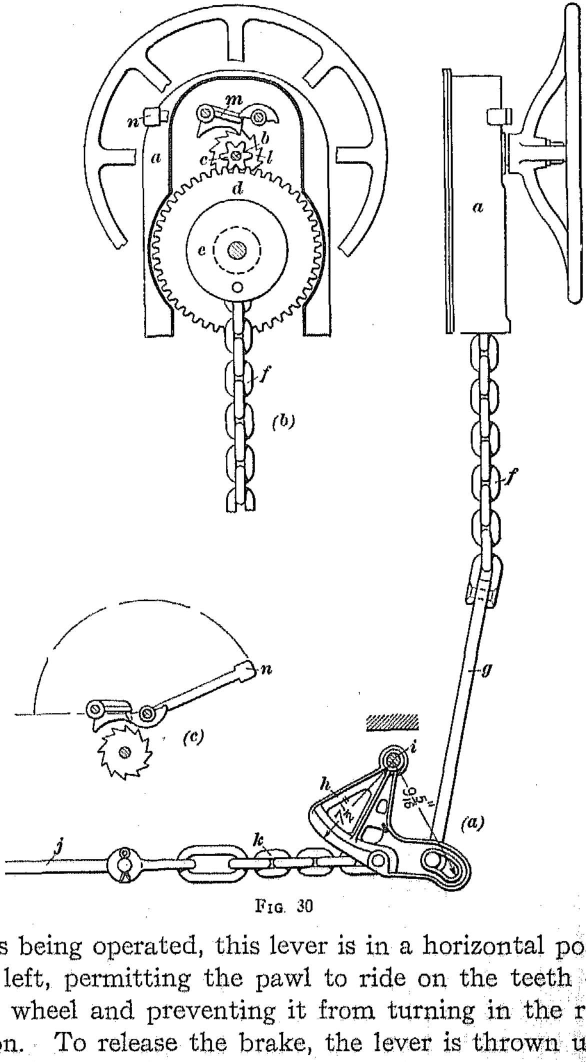


FIG. 30

brake is being operated, this lever is in a horizontal position, on the left, permitting the pawl to ride on the teeth of the ratchet wheel and preventing it from turning in the reverse direction. To release the brake, the lever is thrown up and over, as shown in (*c*); this lifts the pawl clear of the ratchet

wheel, and permits the hand wheel to turn in the other direction. A movement of the lever between the two extremes permits of a graduated release of the brake.

The bell-crank is a substitute for a sheave that would otherwise have to be applied to the end sill; it also increases the power developed at the chain drum in the ratio of $9\frac{5}{16}$ to $7\frac{1}{2}$, or to $\frac{9\frac{5}{16}}{7\frac{1}{2}}$, or to about 25 per cent.

86. Calculating the Force.—The theoretical force developed by this brake can be calculated as follows:

$$W = \frac{125 \times 22 \times 5.75 \times 9.31}{2 \times 7.5} = 9{,}814 \text{ lb.}$$

in which 125 = assumed pull on rim of brake wheel;
22 = diameter of hand wheel (corresponds to L);
5.75 = gear ratio $\left(\frac{11\frac{1}{2}}{2} = 5.75\right)$;
9.31 = arm of bell-crank, inches;
7.5 = radius, bell-crank chain drum, inches, to center line of chain;
2 = distance from chain drum to center line of chain (corresponds to r).

87. The foregoing shows that to calculate the force developed by a power hand brake, it is only necessary to multiply the force applied to the hand wheel by its diameter, and then by the gear ratio, and divide by the radius of the chain drum. The gear ratio can be calculated either by dividing the number of teeth in the chain drum gear by the number in the pinion gear, or it can be obtained from the pitch diameter when given. Thus the gear ratio of the Peacock brake is $\frac{49}{7}$ or 7 and that of the Ajax brake is $\frac{46}{8}$ or 5.75 if the number of teeth in the chain drum gear is assumed to be 46 and the number in the pinion gear to be 8. When the pitch diameter of the gears is given, the gear ratio is obtained by dividing the pitch diameter of the large gear by the pitch diameter of the small gear as already shown. The determination of pitch diameter is beyond the scope of this lesson paper.

ADJUSTING PISTON TRAVEL

88. When it is found that the piston travel is too long, the position of the live truck levers should first be noted. If both are found to be over their center lines and not at right angles to their rods when the brake is applied, the slack should be taken up equally at both ends by removing the pin in each dead lever guide, Fig. 1, and moving the ends of the levers outwards. After this is done the brake is applied and the piston travel measured. However, if the truck levers are pulled out as far as they will go and then moved back two holes, the travel will be about right.

If the piston travel is still too long with the pins in the dead lever guides taken up to the last hole, the bottom rods should be lengthened one hole. Although not shown in Fig. 1, the bottom rods are usually provided with two or three holes for this purpose. Therefore if a top or bottom rod is badly bent, the piston travel will be too long, and if a rod that is too long is applied, the travel will be too short.

A convenient rule to remember when adjusting the piston travel is that any adjustment of the levers that tends to move the push rod towards the cylinder, shortens the travel, and any adjustment that acts to pull the push rod out of the cylinder, lengthens the travel.

With a brake rigging arranged with the dead lever guides pointing in the other direction from that shown in Fig. 1, the end of the dead truck lever must be pulled in the reverse direction or inwards when adjusting the travel.

LAWRENCE J. LUKENS